COMBUSTION

DEVOTED TO THE ADVANCEMENT OF STEAM PLANT DESIGN AND OPERATION

Vol. 12, No. 3 SEPTEMBER, 1940 25c a copy

McLaughlin Air Service

Sherman Creek Station, New York,
in which million-pound per hour, high-pressure boiler is to be installed

High-Pressure Extension
to Manchester Street Station

•

Efficiency Relationships

•

Sizing of Coal for
Chain- or Traveling-Grate Stokers

•

Steam Generation Trends in 1940

COMBUSTION

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VOLUME TWELVE NUMBER THREE

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FOR SEPTEMBER 1940

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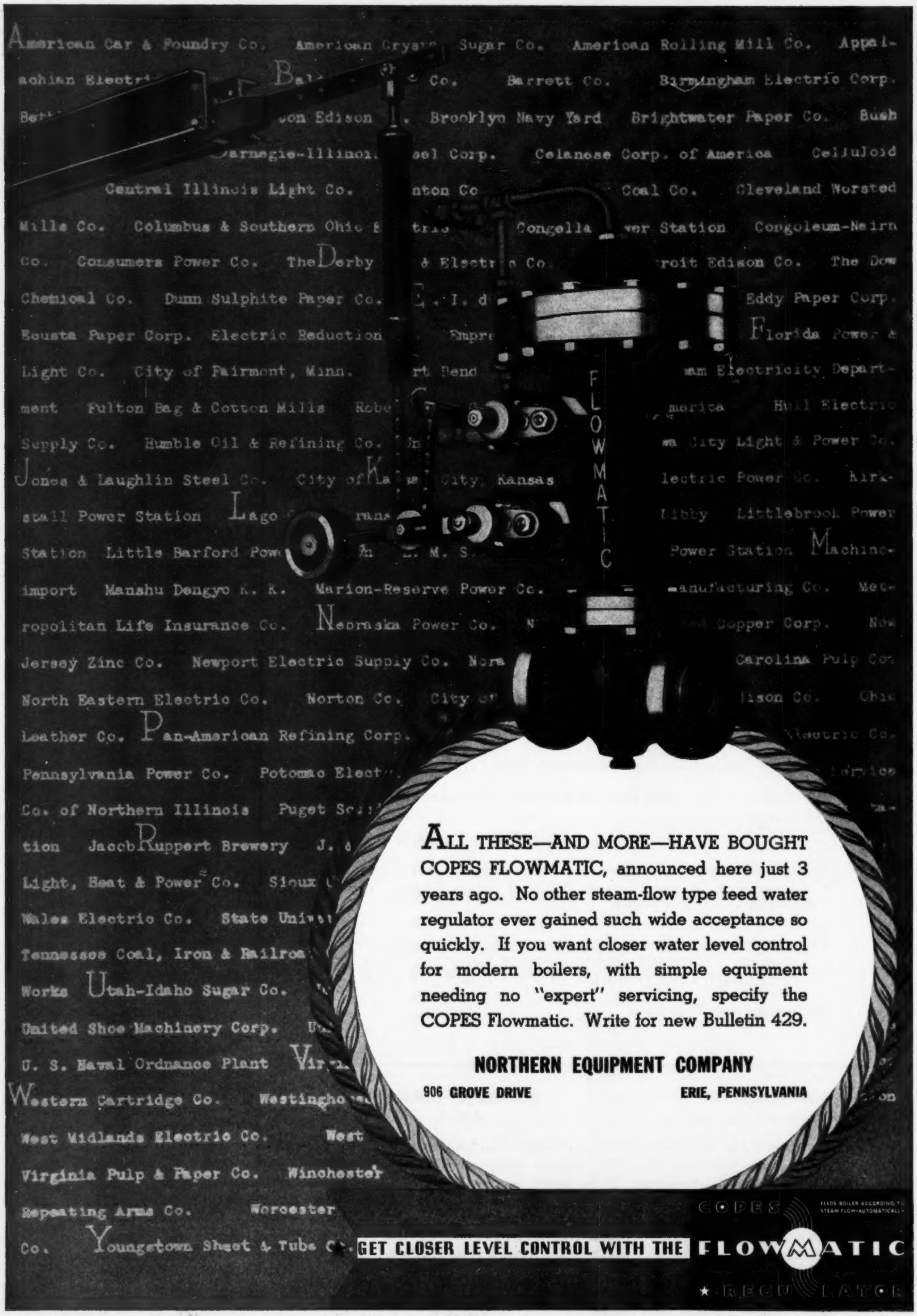
Combustion is published monthly by Combustion Publishing Company, Inc., a subsidiary of Combustion Engineering Company, Inc., 200 Madison Avenue, New York. Frederic A. Schaff, President; Charles McDonough, Vice-President; H. H. Berry, Secretary and Treasurer. It is sent gratis to consulting and designing engineers and those in charge of steam plants from 500 rated boiler horsepower up. To others the subscription rate, including postage, is $2 in the United States, $2.50 in Canada and Great Britain and $3 in other countries. Single copies: 25 cents.

Printed in U. S. A. Publication office, 200 Madison Avenue, New York. Issued the middle of the month of publication.

Member, Controlled Circulation Audit, Inc.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EDITORIAL

Some Probable Effects of Minimum Coal Prices

With announcement by the Government that the new schedules of minimum bituminous coal prices will definitely go into effect on October first, many consumers are pondering the effect of these new prices on the selection and cost of coal to meet their individual requirements.

The schedules are so voluminous and their application so intricate as to make impossible any definite predictions, even by those having coal purchasing background. It will be recalled that a few of these schedules, applying to some representative localities and consuming markets, were reproduced in COMBUSTION last month. So many factors are involved that the probable general effect is debatable. However, certain facts point to some reasonably plausible deductions.

Top minimum prices, applying to the better grades, are not materially different from present market quotations, but the schedules provide for a general rise in the level of prices affecting the lower grades. That is, the spread between minimum prices for the better and the poorer coals will be substantially decreased. This is particularly noticeable in regard to some of the middle west coals. In many cases the decrease in differential is disproportionate to the relative values of the coals. Many of those that have been cheapest will be advanced considerably. This situation is likely to create a greater demand for the better grades and thus disturb the production balance. As a result, there will undoubtedly be a marked shift in buying which will bring about a gain in some markets at the expense of others, or a redistribution of markets.

It will be recalled that the Bituminous Coal Act expires in April 1941 and much of the intervening time would logically be required for readjustment in the selection and purchase of coal to meet the new conditions. Whether Congress will see fit to re-enact this legislation in its present or an amended form is a matter that no one can foretell. Should the Act be permitted to lapse the herculean task of arriving at the present minimum prices and the millions spent in this effort will have gone for naught.

Finally, the present level of industrial activity, with still higher levels in the offing, portends a seller's market affecting maximum prices; and the establishment of minimum prices will likely exert a lessened influence, except perhaps as concerns the movement of some of the poorer grades of coal. This would tend to nullify to a large extent the principal motive behind the Bituminous Coal Act, namely, bolstering the coal industry, and might result in reduced pressure on Congress for its re-enactment.

All things considered, for the immediate future the consumer will probably be more concerned with the selection of his coal than the price which he will be compelled to pay for it.

Duplicate or Standard Designs Expedite Preparedness

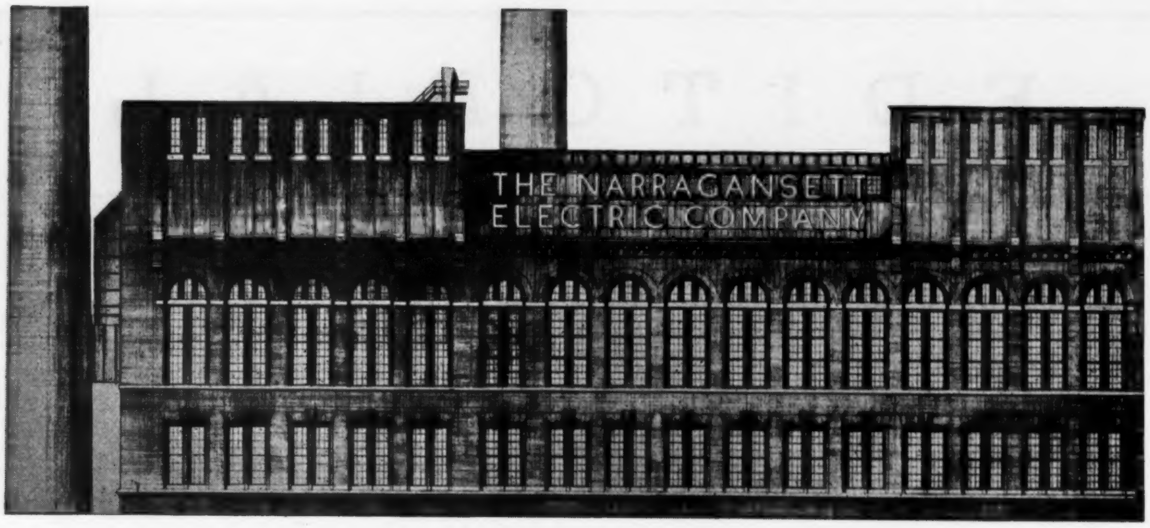
Despite the fact that the Preparedness Program has only recently got under way and will not reach its stride for some months, many orders for power plant equipment have already been placed by firms that are either directly and indirectly concerned with filling government orders. This applies to utilities as well as private industrial plants and the military branches of the Government itself.

The extent to which the necessity for expediting preparedness is likely to affect the selection of steam generating equipment is difficult to assess at this time. Where new plants are to be constructed for munitions manufacture more latitude prevails in the selection of steam conditions, size of units and methods of firing, than where existing plants are to be extended to provide increased facilities.

Final action by Congress in disposing of the much-disputed question of amortization and taxes is likely to influence the selection of equipment, as will also the necessity of securing early productive capacity. As a result, simplicity and quick delivery will be paramount, but these need not be incompatible with good operating economy. All three considerations can be met by selecting more or less standard designs and duplicates of units whose performance has been proved.

Inasmuch as such units involve a minimum of additional engineering work, the time of delivery depends largely upon shop facilities. Equipment manufacturers, who took advantage of the opportunity a few years back to revamp and increase their production facilities, are now in the fortunate position of being able not only to meet the greatly increased volume of orders but also to make prompt deliveries where standardized designs or duplicate units are involved. Under present steel delivery schedules many such units of moderate capacity are now being put out in three to four months, to which time, of course, must be added that required for erection. Larger boilers, especially those to operate under high steam conditions, require a proportionately longer time, as do others where special designs are involved. Should future conditions in the steel industry make for slower shipments of steel the foregoing deliveries would be necessarily extended.

It would appear, therefore, that the provision of adequate steam generating capacity can well keep pace with that of other plant equipment and offers no serious problem in carrying out the Preparedness Program.



Architect's Elevation of Station

High-Pressure Extension to MANCHESTER ST. STATION

By A. S. DAVIS, and GEORGE HIRSCH, Engineers

The Narragansett Electric Company

A description of the high-pressure extension now under construction which will comprise a 430,000-lb per hr, pulverized coal- or oil-fired steam generating unit supplying steam at 1225 lb, 915 F to a 40,000-kw condensing turbine-generator. The new installation replaces eight low-pressure boilers and an old 15,000-kw vertical turbine, in the existing building. Because of the two different steam conditions in the station the unit system of arrangement has been followed and provision made for two additional units to be installed as required.

THE Narragansett Electric Company, of Providence, R. I., has two steam generating stations located about 2000 ft apart, known as the South Street Station and the Manchester Street Station, both of which are approximately half a mile south of the center of the city. A high-pressure installation is now being added to the Manchester Street Station which is ideally located on tidewater at the head of Narragansett Bay and the Providence River which are used for condensing purposes and for the transportation of fuel oil and coal by ocean going steamers.

Before the present high-pressure development was undertaken the station contained 48,000 kw of 25-cycle turbine-generator capacity supplied with steam at 300 lb gage and 600 F from three 120,000-lb per hr boilers and eight 600-hp boilers of a lower pressure. This 25-cycle system which supplies the street railway load direct, is tied with the 60-cycle light and power system through a 20,000-kw frequency-changer set.

A steady growth in the local power and light load and an expected increase in this load because of the highly industrialized locality, made additional generating capacity desirable. As a result of studies made, it was decided to install a 40,000-kw condensing unit and a single boiler of 430,000 lb per hr steam output. Steam conditions of 1215 lb per sq in. pressure and 915 F total steam temperature at the turbine throttle, were selected to obtain the best overall results. The United Engineers and Constructors, Inc., of Philadelphia, were engaged to design and construct this extension.

To make room for this contemplated increase in capacity it was necessary to remove from the station one 15,000-kw vertical turbine-generator and the eight 600-hp low-pressure boilers. By demolition of this obsolete equipment the necessary space requirements were made available for both the high-pressure boiler and the new turbine-generator within the existing building and no extensive changes to the building were necessary.

Because of the different steam conditions involved, the unit arrangement was followed in the new high-pressure extension by eliminating all cross-connections between the new units and the old station. Where feasible during construction, preparation and ground work is being accomplished for two additional unit arrangements of the same capacity. An independent condenser circulating-water system is being constructed

for the new unit, which includes a new intake screen house on the river front, with sufficient capacity to handle circulating water for the present unit and the two contemplated units. All circulating-water pumps will be located in the screen well house with the intake screens and for the present unit a 54-in. diameter circulating-water pipe line is being installed and improvements to the existing discharge tunnel are being made.

An existing brick stack which previously served the eight old boilers is being connected with the new boiler. This stack is located outside and adjacent to the present building structure.

The generator leads will be laid directly to a bus structure in a nearby building where all of the 60-cycle main switches are located. These main switches are remote controlled from the main operating control board within the power plant proper. In the building containing this bus structure there is also located the dispatching center for the Rhode Island territory.

Two new 42,500-barrel fuel oil tanks will provide storage for fuel oil. These tanks are directly connected through a pipe line with the oil tanks at the South Street station, thus making the tank capacity at both stations available for either of the generating plants.

Although the new boiler will be equipped for burning coal in addition to the oil, no special coal unloading and handling equipment is being installed at this time. The present coal handling equipment for the 25-cycle plant will be used for supplying the high-pressure plant with coal. A coal bunker is being installed with capacity sufficient to operate the boiler at full load for thirty hours. By this installation either fuel oil or coal are available for the operation of the high-pressure unit. Sufficient yard space for coal is available for the storage of 30,000 tons.

Steam Generating Unit

The boiler, of 430,000 lb per hr steam generating capacity, is of the three-drum Combustion Engineering Company bent-tube type with a convection-type superheater located in the second pass, and a continuous-loop horizontal fin-tube economizer below the rear drum within the boiler casing. Two vertical-shaft air preheaters of the Ljungstrom regenerative type are located on the gallery above the operating floor. The furnace which has a volume of 21,500 cu ft and is of the dry-bottom type, is completely surrounded by fin-tube water walls. It is equipped with six burners of the horizontal combination type designed for burning either pulverized coal, or fuel oil by means of mechanical atomizers. Pulverized coal will be supplied by three C. E.-Raymond bowl mills located as shown in the boiler cross-section.

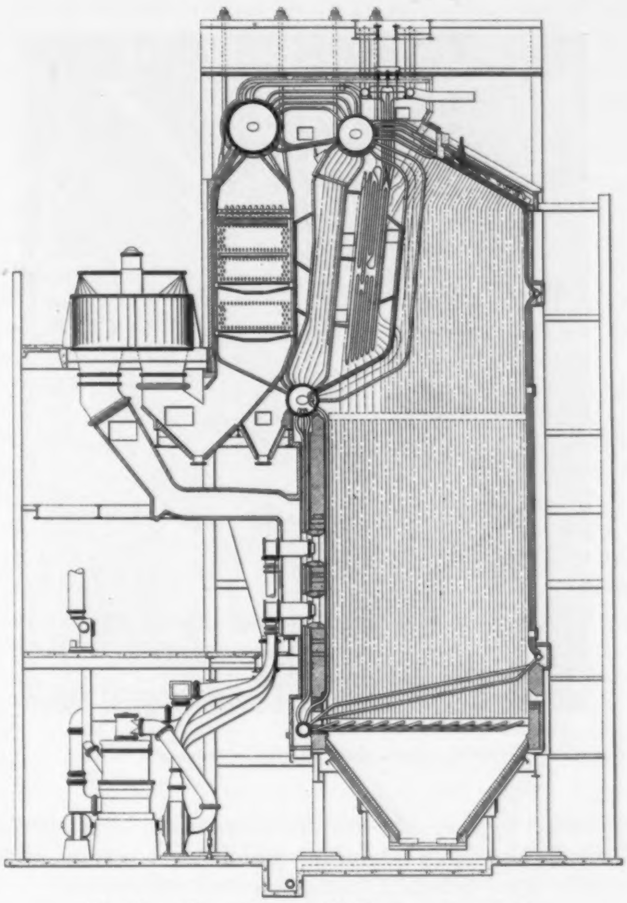
Distribution of the heat-absorbing surfaces of the unit is as follows:

Boiler	7,775 sq ft
Water walls	7,153 sq ft
Superheater	9,170 sq ft
Economizer	15,825 sq ft
Air heaters	60,980 sq ft

The feedwater treatment is arranged with an external softener system feeding the treated makeup water to an evaporator, but direct feed of chemicals to the boiler drum is also possible if found necessary.

The boiler is designed for a steam pressure of 1350 lb per sq in. or 1225 lb at the superheater outlet nozzle. A Leeds & Northrup full automatic combustion control will control operation at all loads when burning either pulverized coal or fuel oil. The superheater is designed for 915 F total steam temperature at full load, and an automatically controlled bypass damper is located in the upper part of the boiler to bypass a portion of the gases around the superheater when necessary. This damper is regulated by a Leeds & Northrup automatic superheater control.

Air to the furnace and the products of combustion will be handled by two Sturtevant vane-controlled



Section through new high-pressure unit

forced-draft and two induced-draft fans each having two-speed induction motor drives.

The expected operating characteristics of the unit are:

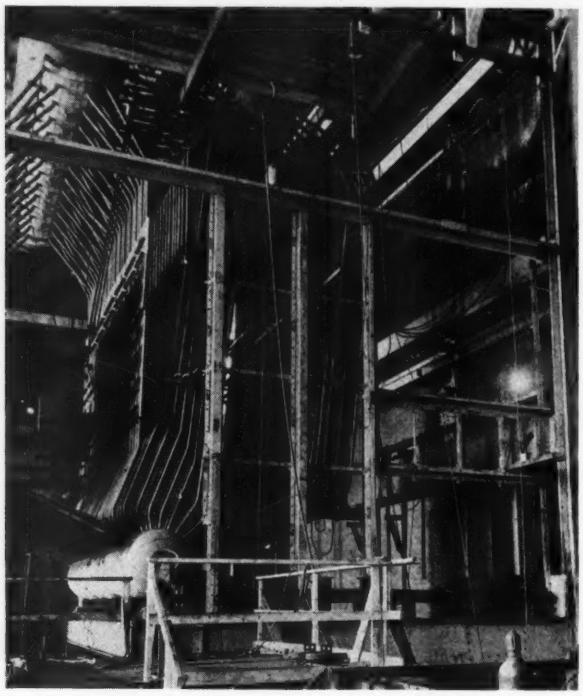
Steam generated (rated capacity)	430,000 lb per hr
Steam pressure at superheater outlet	1,225 lb per sq in.
Drum pressure	1,285 lb per sq in.
Temperature of steam at superheater outlet	915 F
Temperature of feedwater to economizer	400 F
Temperature of air leaving air heater	493 F
CO_2 at economizer outlet	15 per cent
Total draft loss through furnace, boiler, superheater, air heater	11.67 in. of water
Temperature of gases leaving air heater	288 F
Furnace combustion rate	24,500 Btu per cu ft per hr
Boiler efficiency at 350,000 lb per hr	88.9 per cent

High-Pressure Turbine-generator

The new turbine-generator is of 40,000 kw capacity and built by the General Electric Company. The high- and low-pressure cylinders are in tandem and directly connected to a 50,000-kva 3-phase, 60-cycle, 11,500-volt, 3600-rpm hydrogen-cooled generator. The turbine will operate at a throttle steam pressure of 1215 lb per sq in. and at a throttle steam temperature of 915

F. The generator is designed to give the full 50,000 kva rating at unity power factor. There will be four extraction points on the turbine, located at the 7th, 10th, 14th, and 19th stage, respectively. The exciter and pilot exciter will be driven through a reducing gear on the outboard end of the generator shaft.

The turbine is equipped with a quick load-dumping device so that if certain switches on some of the high-tension lines should open from overload the unit will dump its entire load in about fifteen cycles and will then automatically pick up a predetermined 3000 kw



Construction view of high-pressure unit

in order to keep the turbine rotor from overheating. This device is to maintain stability on certain high-tension lines radiating from the power station bus.

The hydrogen cooling system is arranged for maximum hydrogen pressure inside the generator for emergency overload capacities.

Auxiliaries

All of the auxiliary equipment in connection with the boiler and turbine-generator are electrically driven with the exception of the stand-by boiler feed pump which is steam-turbine driven at 1225-lb pressure and saturated temperature.

The auxiliary main leads are taken from two places, one directly from the leads of the generator and the other from the generator bus which is backed up by other sources of power supply.

The feedwater heating will utilize steam from the four extraction points of the turbine in a regenerative cycle, raising the boiler feedwater from condenser temperatures to approximately 400 F at the inlet to the economizer. The fourth-point extraction heater is of the direct-contact type in which the steam intermingles directly with the water. The third bleed point is supplied with steam slightly above atmosphere at full load and the heater is of the closed or tubular type in which the heat is transmitted by conduction. A direct-contact type heater is used for the second bleed point and the first bleed point is equipped with a tubular heater designed for operating at high pressures, as the boiler feed pump takes its suction from the second bleed-point heater, thus putting the first bleed-point heater on the boiler-feed pump discharge.

The evaporator takes its steam from the second bleed point and discharges to the heater on the third bleed point. The boiler level is automatically controlled and the level of the water in the direct contact heaters are also automatically controlled, as well as the quantity of water entering the heating system from the condenser hotwell, by a Bailey three-point automatic regulating system.

There is a condensate storage tank of 500,000 lb capacity, installed to take care of the surge between the condenser hotwell discharge and the input to the boiler. This tank also acts as storage for distilled water at all times for emergency purposes.

Condensing Equipment

The surface condenser is of the single-pass Ingersoll-Rand type with 28,500 sq ft of tube surface. The tubes are of aluminum-brass, which after extensive research and tests proved to be best adapted to meet the existing conditions. They are 24 ft long, $^7/_8$-in. diameter and 18 gage. The water boxes are streamlined to minimize turbulence of the circulating water. It has a duplicate set of steam-jet air pumps, duplicate condensate pumps, and two circulating-water pumps. The circulating-water pumps are located at the river in the screen-well house and are of the vertical submerged propeller type with the motors above the high-water mark.

All steam and water piping in the plant where possible will be of welded construction and selected to conform with the latest practice.

The company has retained the following consultants in addition to the previously mentioned engineering organization.

Metallurgy	Prof. A. E. White
Boiler Feed Treatment	Shepard T. Powell
Architectural Details	Paul P. Cret

Principal Equipment for Manchester Street Station Extension

Company—*The Narragansett Electric Company*
Location—Providence, Rhode Island
Service—Electric light and power
Engineers and Constructors—*United Engineers & Constructors, Inc.*

BOILER AND AUXILIARIES

Boiler

One three-drum bent-tube type, 8285 sq ft boiler heating surface; water walls 6790 sq ft. Diameter of drums: front 42 in., rear 60 in., bottom 36 in. Boiler constructed for 1350 lb per sq in.; capacity 430,000 lb per hr actual evaporation. One Elesco convection type 2 in. O.D. tubing, 10,000 sq ft effective heating surface designed for 915 F total steam temperature at 430,000 lb per hr. Temperature automatically controlled by operating a damper to bypass a portion of boiler gases. One Elesco fin-tube economizer with 16,650 sq ft effective heating surface designed to raise the feedwater temperature 100 deg F with a flow of 430,000 lb per hr.—*Combustion Engineering Company, Inc.*

Soot Blowers

Twelve soot-blower elements of the revolving type and four elements of the stationary type. Steam to be obtained from boiler drum and reduced to 400-lb operating pressure. Arrangements are to be made to install telescopic units in the future if found necessary.—*Diamond Power Specialty Company.*

Air Preheaters]

Two Ljungstrom regenerative vertical shaft type each with 29,400 sq ft total heating surface, designed to raise the air temperature 400 deg F when boiler is operating at full output and burning pulverized coal or fuel oil.—*The Air Preheater Corporation.*

Pulverizers

Three C.E.-Raymond bowl mills of the unit type with integral exhauster. Each mill has a capacity of 18,900 lb per hr.—*Combustion Engineering Company, Inc.*

Fuel Burners

Six type RF horizontal combined pulverized coal and fuel oil burners with *Coen* tips for oil. Each burner has a capacity of 6200 lb of coal or 6000 lb of oil per hr.—*Combustion Engineering Company, Inc.*

Forced-Draft Fans

Two double-inlet vane-control Turbo-vane type, direct connected to two-speed squirrel-cage motors. The fans have a combined capacity of 130,000 cfm at 9.2 in. of water at 100 F.—*B. F. Sturtevant Company, Inc.*

Induced-Draft Fans

Two double inlet vane-control fans direct-connected to two-speed squirrel-cage motors. The fans have a combined capacity of 237,000 cfs at 22 in. of water at 320 F.—*B. F. Sturtevant Company, Inc.*

Combustion Control

Complete automatic combustion control for boiler and its auxiliary equipment.—*Leeds & Northrup.*

Steel Plate Work

A boiler breeching, one forced-draft duct and one induced-draft duct to include all the necessary clean-out doors, dampers, and damper-operating mechanism.—*James Russell Boiler Works.*

FEEDWATER SYSTEM

Boiler Feed Pumps

Two 5-in. 7-stage centrifugal, horizontal, single-suction type, one direct-connected to a *Westinghouse* constant-speed motor drive and the other to a *Westinghouse* high-pressure single-stage steam turbine. Total dynamic head 3800 ft when boiler is operating at full output; speed 3600 rpm for normal conditions.—*Ingersoll-Rand Company.*

Feedwater Regulators

Three-point control on boiler, each heater and condenser hotwell. The controls on boiler and each heater will be connected to a master controller that will keep system in a stabilized condition during changes of flow.—*Bailey Meter Company.*

Extraction Heaters

One first-point bleed closed heater, one third-point bleed closed heater, and one heat exchanger or drain cooler.—*Griscom-Russell Company.*

One second-point bleed, direct-contact heater and one fourth-point bleed, direct contact heater.—*Foster Wheeler Corporation.*

Evaporator

One single-effect evaporator ordinarily to produce vapor at about 22 lb per sq in. gage, using steam at about 92 lb. Coil surface 360 net sq ft, tubes 1.0 in. O.D.—*Griscom-Russell Company.*

TURBINE-GENERATOR AND AUXILIARIES

Turbine-Generator

One 40,000-kw 3600-rpm tandem-compound, double-flow condensing unit arranged for four-point steam extraction with a turning gear and a special quick-opening throttle-trip valve. Steam pressure 1215 lb gage, 915 F total temperature. Generator hydrogen cooled, 3-phase, 60 cycles, 11,500 volts, 0.8 power factor with a gear-connected exciter and pilot exciter. One 165-kw, 250-volt, 1800-rpm exciter.—*General Electric Company.*

Surface Condenser

One 28,500-sq ft condenser of welded shell construction, but with cast-iron water boxes. Single-pass type, 24 ft active tube length and with two circulating water pumps having a combined capacity of 60,000 gpm.—*Ingersoll-Rand Company.*

Water Valves

Motor operated intake valves.—*Chapman Valve Manufacturing Company.*

Intake Water Screens

Two vertical travelling type.—*Link-Belt Company.*

De-Sliming Process

Chlorinaters and accessories to inject chlorine gas into condenser-circulating water piping.—*Wallace & Tiernan Products, Inc.*

Generator Oil Circuit-Breakers

H-Type-3000 amp 15 kv.—*General Electric Company.*

STATION SERVICE

Fan motor controls.—*General Electric Company.*
Motor drive on auxiliaries.—*Westinghouse Electric & Mfg. Co.*

INSTRUMENTS

Steam flow meters.—*Bailey Meter Company.*
CO_2 recorder.—*Leeds & Northrup Company.*
Feedwater flow meters.—*Builders Iron Foundry.*

MISCELLANEOUS EQUIPMENT

Ash removal system.—*United Conveyor Corp.*
Piping.—*Grinnell Company.*
Boiler safety valves.—*Consolidated Safety Valve Co.*
Boiler stop check valve.—*Schutte & Koerting.*
Boiler blow-off valves.—*Yarnall Waring Company.*
High-pressure steam gate valve.—*Chapman Valve Mfg. Co.*
Bleed check valves.—*Atwood & Morrill Co.*
Condensate booster pumps.—*Ingersoll-Rand Co.*
Lubricating oil-purification equipment.—*Turbine Equipment Co.*
Fuel-oil pump.—*Quimby Pump Company.*
Fuel-oil heater.—*Alco Products, Inc.*

Economizer Corrosion

A reader inquires as to the factors causing economizer corrosion and how it may be avoided.

Internal corrosion of economizer tubes may be due to entrained oxygen in the feedwater or to low hydrogen ion concentration (pH). Deaerating heaters are employed to remove entrained oxygen, but in some cases a supplementary treatment of sodium sulphite is used to remove any traces of oxygen that may get past the heater. Chemical means are commonly employed to control the proper pH of the feedwater passing through the pumps and economizers. However, in an increasing number of plants from 1 to 2 per cent of the boiler water is recirculated through the feedwater system to build up the pH value. This decreases the amount of chemical treatment, with a view to lessening concentration in the boiler.

External corrosion of economizers is caused by the presence of sulphur compounds in the products of combustion if cooled to or below their dew points, the sulphur combining with moisture to form sulphurous and sulphuric acids which will attack the steel. Conditions favorable to corrosion are high sulphur fuels and low feedwater temperature when the unit is operating at light load or is banked for long periods.

Such corrosion is more likely to take place at the colder end of the economizer, particularly if the feedwater temperature is low, or if the gas velocity is low and there are pockets of more or less stagnant gas having a low final temperature.

Organic and pyritic sulphur present in solid fuel oxidize to sulphur dioxide (SO_2) and sulphur trioxide (SO_3), which are to be found as such in the products of combustion. The proportions of these compounds will vary with different fuels and to some extent with the methods of firing. It is common, however, to find about one-third of the sulphur in the flue gas in the form of SO_3, the balance being SO_2. At a temperature below about 620 F the SO_2 combines with water vapor present in the flue gas. This water vapor comes from three sources, namely, moisture in the fuel, oxidation of hydrogen in the fuel and moisture present in the air used for combustion. Soot-blowing, particularly with low-pressure steam, may add considerable moisture to the flue gas, and, where low metal temperatures are encountered, a film of moisture may accumulate on the tubes. When the dew point is reached, sulphuric acid begins to condense and upon further cooling water vapor is absorbed and corrosive conditions are set up.

It has been suggested that the sticky deposit often found on economizer tubes of oil-fired boilers may act as a binder for the corrosive agents in the products of combustion and thus accelerate tube deterioration.

The destructive effects of mechanical abrasion may also be considered as it is not uncommon for misaligned soot-blowers to cause cutting of the tubes.

In general, external corrosion of economizer tubes is more likely to be experienced among low-pressure units, where the feedwater temperature is lower and operating conditions are less favorable. Geographical location or other considerations may necessitate the use of high-sulphur fuel and load conditions determine the banking periods or operation at light loads, but it is usually possible to raise the feedwater temperature. The recommended minimum, where high-sulphur fuels are burned, is about 190 F, although a higher temperature is desirable.

Efficiency Relationships

By LYBRAND SMITH

Capt., U. S. Navy, Head of Research Branch, Bureau of Ships

Because of the confusion that often exists in discussions of, or references to efficiencies, where the basis of their computation is not given or symbols are not uniform, the author explains the relationship between the several efficiencies commonly employed in steam engineering, and offers a consistent set of symbols to facilitate comparisons.

EFFICIENCY is a misleading word. Many misunderstandings and useless arguments[1] result from failure to state the operation by which efficiency is computed.

Familiar as we all are with the formulas for computing various efficiencies, this situation would not seem possible had not so many fruitless arguments been witnessed. Further consideration of the matter indicates that possibly the situation is caused not only by formulas being scattered widely through reference books, but also by the use of different symbols for the same quantities in different formulas. These two points alone are enough to obscure the relationships which exist between efficiencies.

The present notes are intended merely to illustrate certain relationships between efficiencies frequently used in steam engineering, by assembling a few standard formulas in one article, and writing them in consistent symbols.

Symbols

The following are the symbols which will be used. There is no particular merit in these symbols except that they are consistent in this paper, are more or less mnemonic, and, where possible, conform to symbols used in such standard references as Keenan and Keyes Steam Tables (a)[2] or Mark's Handbook (b).

E = efficiency in general
E_c = Carnot cycle efficiency
E_r = Rankine cycle efficiency
E_n = Engine efficiency
E_t = Thermal efficiency
T = highest absolute temperature in a cycle = deg F. + 459.69
T_o = lowest absolute temperature in a cycle = deg F. + 459.69
h_f = enthalpy of liquid (fluid) at condenser conditions, Btu per lb
h_g = enthalpy of highest condition steam (gas), Btu per lb
h_x = enthalpy of exhaust, after isentropic expansion from h_g, Btu per lb
h_{xa} = enthalpy of exhaust, after *actual* expansion from h_g, Btu per lb
q_a = heat per lb of steam *actually* transformed into work by an engine. (See discussion of measurement of this quantity in the section on "Engine Efficiency.")
q_b = heat per lb of steam furnished by boiler = $(h_g - h_f)$
q_m = heat per lb steam which is maximum possible to transform into work = $(h_g - h_x)$
N_r = steam (water) rate of ideal engine, in lb per hp hr
N_a = steam (water) rate of actual engine, in lb per hp hr
H_a = heat converted to work by actual engine, in Btu per hp hr
H_b = heat supplied to actual engine in Btu per hp hr

Efficiency in General

The general definition of efficiency is:

$$E = \frac{\text{Output}}{\text{Input}} \tag{1}$$

This has no special limitations except that *output* and *input* must be expressed in the same units, so that E is dimensionless; and *output* must be less than *input* so that E is less than unity; otherwise we would have perpetual motion. The different varieties of *efficiency* arise because, for purposes of convenience, it is often desirable to use different items for output and input. Misunderstandings sometimes arise from lack of clearness as to just what these items should be.

Carnot Cycle Efficiency

Most modern textbooks give the impression that Carnot was a hazy theorist dealing with a hypothetical perfect gas. Nothing could be further from the truth. His classic paper (c) reveals him as a practical, clear

[1] Letters in parentheses refer to bibliography at the end of this paper.
NOTE—Opinions expressed in this article are personal ones of the writer and not necessarily official opinions of the Navy Department.

thinking engineer dealing principally with steam but open-mindedly examining the uses of other vapors, such as that of mercury, and of gases, such as air and carbon dioxide.

The formula which modern textbooks now use to give the efficiency of the Carnot cycle apparently was based on William Thompson's (Lord Kelvin's) paper of 1848 (d). In this paper Kelvin said:

> "The characteristic property of the scale which I now propose is that all degrees have the same value; that is, that a unit of heat descending from a body A at the temperature $T°$ of this scale to a body B at the temperature $(T - 1)°$, would give out the same mechanical effect, whatever be the number T. This may justly be termed an absolute scale, since its characteristic is quite independent of the physical properties of any specific substance."

Based on Kelvin's idea of the absolute temperature scale, we get for the Carnot Cycle Efficiency the familiar formula:

$$E_c = (T - T_o)/T = 1 - T_o/T \qquad (2)$$

Notice that the reason one can use absolute temperatures this way for *output* and *input* of energy is that the energy content at any temperature T is assumed as kT, where k is a proportionality factor which remains constant over the whole temperature range from absolute zero up. Hence k cancels out of equation (2) and T is used as an indicator of the amount of energy *above the level of absolute zero.*

A *perfect gas* would behave this way, but real substances only approximate such behavior; as, for example, a study of a Mollier Diagram or a Temperature-Entropy Chart for steam will show. Hence, in practical cases we cannot use T as an indicator of energy, but have to employ tables to find the energy content corresponding

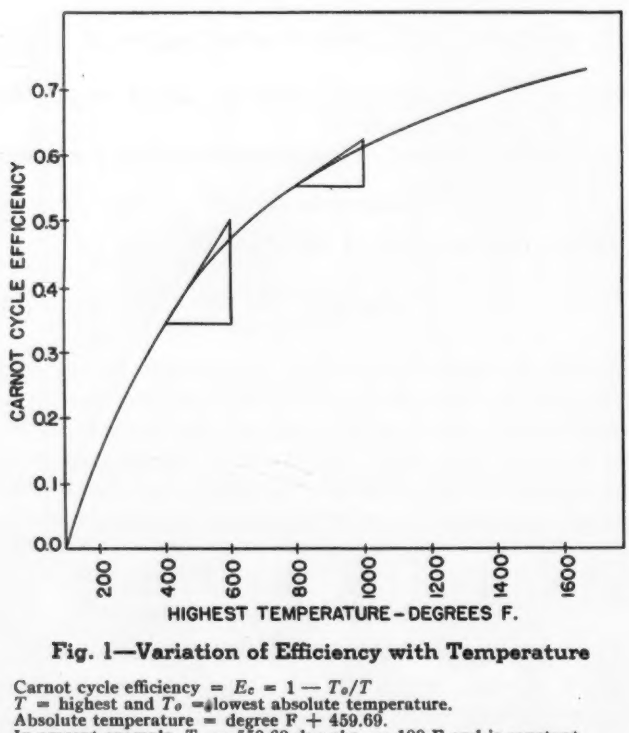


Fig. 1—Variation of Efficiency with Temperature

Carnot cycle efficiency $= E_c = 1 - T_o/T$
T = highest and T_o = lowest absolute temperature.
Absolute temperature = degree F + 459.69.
In present example, T_o = 559.69 deg abs. = 100 F and is constant.
$\partial E/\partial T = T_o/T^2$,
At 400 F, $\partial E/\partial T = 7.67 \times 10^{-4}$
At 800 F, $\partial E/\partial T = 3.52 \times 10^{-4}$

to various values of T. Therefore, we might as well ignore the T and arrange our tables in some more convenient form. This is exactly what has been done in such tables as those of Keenan and Keyes. Notice, moreover, that in the practical tables we do not measure energy above a level of absolute zero, but measure from an arbitrary reference value. In the case of the steam tables, we choose to assert that the enthalpy is zero for saturated liquid water at 32 F, and measure the energy changes from that arbitrarily selected point.

For these reasons, one does not calculate Carnot Cycle Efficiency very often; but it is well to remember that this efficiency is fundamentally of the same nature as the other efficiencies to be discussed, and is related to them as representing the limit which the efficiencies of heat engines can approach but not surpass.

Fig. 1 shows an example of the variation of Carnot Cycle Efficiency with temperature; and illustrates that, although the efficiency continues to increase while the temperature range increases, yet there is a *law of diminishing returns.*

Rankine Cycle Efficiency

Turning from an ideal perfect gas to a real substance, such as steam, Rankine (e) gave a definition of efficiency which would apply to either. His definition is as follows:

> "The *efficiency* of an engine is the proportion which the energy permanently transformed to a useful form by it, bears to the whole energy communicated to the working substance."

This definition would apply whether we were computing the theoretical maximum efficiency obtainable from steam under certain conditions, or only some fraction thereof, as in actual practice. As a matter of convention, however, the former meaning is customarily understood when Rankine Cycle Efficiency is mentioned; and that is the meaning which will be used in this paper. On this basis, we get as formulas for Rankine Cycle Efficiency, where the complications of bleeding or reheating are not involved:

$$E_r = q_m/q_b = (h_g - h_x)/(h_g - h_f) \qquad (3)$$

Engine Efficiency

Rankine Cycle Efficiency shows the maximum possible efficiency obtainable from steam working between two different states, as applied to a perfect or ideal engine. Another measure of efficiency is needed to indicate the merits of the engine or mechanism used to transform the theoretical possibility into actuality. This measure is called the "Engine Efficiency" and is normally computed by the following equations:

$$E_n = q_a/q_m = q_a/(h_g - h_x) \qquad (4)$$

The value of q_a depends in part upon what is considered *the engine.* If we are concerned with shaft horsepower delivered to a propeller, then the turbine plus its gear may be the engine, and q_a is computed from measurements of shaft horsepower. If we are concerned only with action inside the turbine, then q_a is involved with horsepower delivered by the turbine rotor, and in this case q_a must equal $(h_g - h_{xa})$. In this special case, E_n is sometimes called "Internal Efficiency," and equals $(h_g - h_{xa})/(h_g - h_x)$.

A measure of Engine Efficiency which is superficially different from Equation (4) is:

$$E_n = N_r/N_a \tag{5}$$

According to Keenan and Keyes (f), 2544.1 Btu are equivalent to one horsepower-hour. Other authorities use slightly different values, but in round numbers let 2544 Btu equal one horsepower-hour.

Then the pounds of steam necessary to generate one horsepower-hour in an ideal engine would be:

$$N_r = 2544/q_m \tag{6}$$

and the pounds of steam necessary to generate the same power in an actual engine would be:

$$N_a = 2544/q_a \tag{7}$$

Hence,

$$N_r \times q_m = 2544 = N_a \times q_a \tag{8}$$

Therefore, by equations (4) and (8)

$$E_n = q_a/q_m = N_r/N_a \tag{9}$$

As in the case of q_a, N_a depends in part upon where the power is measured. The same number of pounds of steam per hour may pass through the turbine, but the horsepower output is different if it is measured before or after the reduction gears.

Thermal Efficiency

Rankine Cycle Efficiency shows what is theoretically possible to get out of steam. Engine Efficiency shows how nearly a particular engine realizes the theoretical possibilities. Another measure is needed to show what is actually got for what is actually put in. This measure we call "Thermal Efficiency" and define it as follows:

$$E_t = H_a/H_b \tag{10}$$

Now of necessity (see Equation (8))

$$H_a = q_a \times N_a = 2544 \text{ Btu} \tag{11}$$

and

$$H_b = q_b \times N_a \tag{12}$$

Hence E_t can be expressed in a different form because by Equations (10), (11) and (12),

$$E_t = q_a/q_b = 2544/q_b \tag{13}$$

In the case of shore power plants *plant thermal efficiency* is defined as 2544/(Btu fired per hp hr), or 3413/(Btu fired per kw hr). If the horsepower-hours or kilowatt-hours used in the denominator are the total generated, the *gross* efficiency is obtained. If only the power sent out of the plant is used, the *net* efficiency is obtained. Obviously, these efficiencies are analogous to that defined by Equation (13).

Sometimes the question is asked: Do we save much fuel if we gain a few per cent better plant thermal efficiency? If efficiencies are low, the answer is decidedly yes; although there is a law of diminishing returns. For example, increasing plant thermal efficiency from 15 to 20 per cent means a decrease of 25 per cent in fuel consumption. An increase of efficiency from 20 to 25 per cent means a decrease of 20 per cent in fuel consumption. The general relationships are shown graphically in Fig. 2.

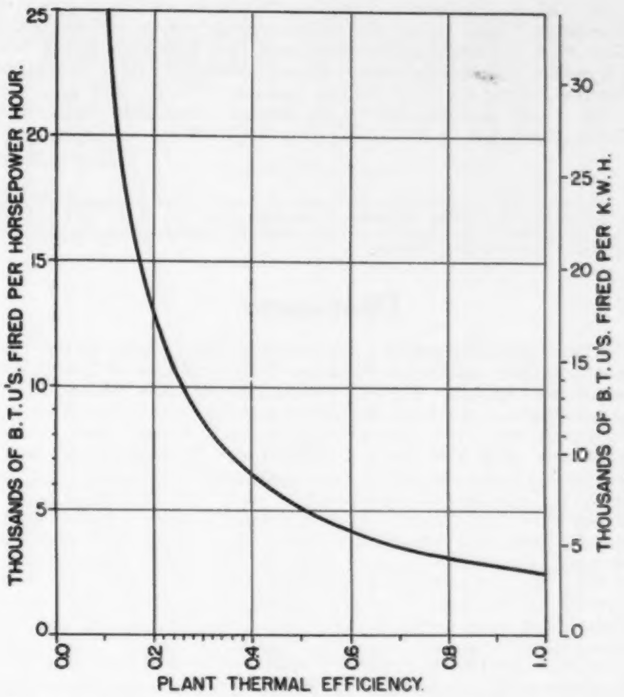


Fig. 2—Reduction in fuel consumption with increased efficiency

PLANT THERMAL EFFICIENCY = 2544/(Btu fired per hp hr)
= 3413/(Btu fired per kw hr)

Relationship of Three Efficiencies

Turning to Equations (3), (4) and (13) it will be seen that, because

$$(q_m/q_b) \times (q_a/q_m) = (q_a/q_b) \tag{14}$$

Rankine Cycle, Engine and Thermal Efficiencies are related as follows:

$$E_r \times E_n = E_t \tag{15}$$

Conclusions

If there be any important point in this paper it is not the specific relationships presented. It is the point that when talking about, writing about, or conferring about efficiencies of any kind from those of boilers to those of diesel engines we should insist on an unequivocal statement of the formula or operation which defines the particular efficiency under discussion. In short, we should always remember—*Efficiency* may be a misleading word.

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The following comments on that part of the foregoing discussion, dealing with the "Carnot Cycle Efficiency," are offered by Dr. E. F. Leib and, at Captain Smith's suggestion, are printed with the original article for the convenience of the reader—Editor.

Discussion

When Carnot's pamphlet "Sur la Puissance motrice du feu" appeared in 1824, neither the First nor the Second Law of Thermodynamics was known. He believed that in the cycle process, which he investigated, the same amount of heat which is given off by the receptacle with the higher temperature is turned over to the receptacle with the lower temperature. Equivalence of work and heat was introduced into thermodynamic reasonings when, in 1842, the mechanical equivalent of heat was calculated by R. Mayer from Gay-Lussac's experiments on the specific heat of gases and, in 1843, was found from direct experiments by Joule, who measured the heat of friction in flow through pipes. It was not until 1847, that the Law of Conservation of Energy, which is the First Law of Thermodynamics, was enunciated by Helmholtz. Since then, it has been recognized that in a work producing process, such as the Carnot cycle, an amount of heat equivalent to this work disappears and that less heat is turned over to the receptacle with the lower temperature than was given off by the receptacle with the higher temperature.

But still the concept of absolute temperature was based on observations of the contraction of gases, and the formula given by Carnot for the efficiency of his cycle process could be proved true only for such substances in which the energy was dependent on the temperature alone (it is not necessary that the energy be proportional to absolute temperature). Only after the Second Law of Thermodynamics was found by Clausius in 1850 was it possible to establish the absolute temperature as a universal property independent of any particular substance. It is one of the consequences of the Second Law that the maximum efficiency of any process, operating between the temperatures T and T_o, in which work is produced by conversion of heat, is given as

$$E = \frac{T - T_o}{T}$$

and is absolutely independent of the properties of the working substance. The Carnot cycle is only one among many others which yield this maximum of efficiency. Each cycle process in which all heat is transferred to the working substance at the highest temperature (T), and in which all heat is given off by the working substance at the lowest temperature (T_o) of the cycle, yields the same efficiency as the Carnot cycle.

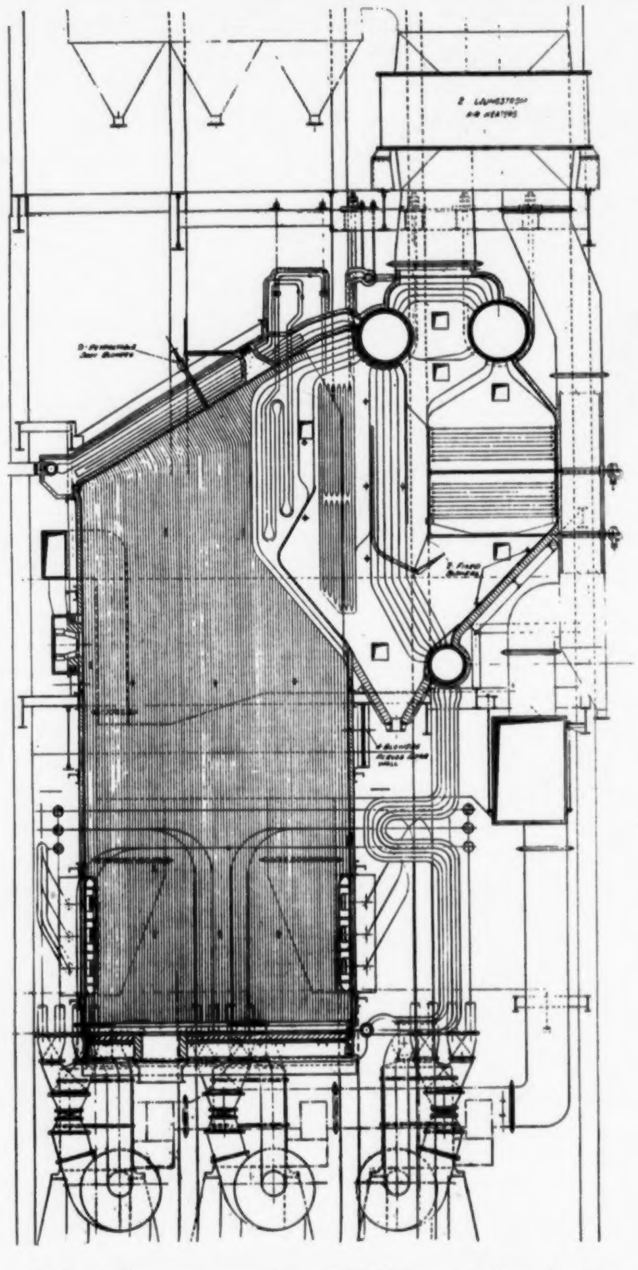
E. F. Leib
Combustion Engineering Co., Inc.

Million-Pound per Hour Boiler Tops Sherman Creek Station

The latest extension in the construction program of the Consolidated Edison Company of New York, Inc., is the topping of the Sherman Creek Station, located at 201st Street and the Harlem River. This 200-lb station, placed in operation in 1913 and increased in capacity from time to time, contains six turbine-generators aggregating approximately 140,000 kw.

This topping installation will consist of a million-pound per hour steam generating unit, designed for 1775 lb pressure and supplying steam at 1600 lb, 955 F to a 50,000-kw, 3600-rpm turbine-generator exhausting to the present 200-lb station header.

The steam generating unit, for which contract was let to Combustion Engineering Company on August 26, will be of the three-drum bent-tube type with Elesco superheater and economizer, two Ljungstrom regenerative air preheaters and a steam washer in the steam drum. It will be tangentially fired with pulverized coal from three C-E Raymond bowl mills and have a continuous slag-drip furnace bottom. In addition to the twelve coal



Preliminary cross-section of million-pound per hour, high-pressure unit

burners, three in each corner, there will be two oil burners in each corner, between the coal burners, and four horizontal oil burners, located in the front upper wall. The latter are for maintaining performance with oil comparable to that when burning coal.

Plain bifurcated tubes will be employed for the front, rear and side furnace walls and the roof and bottom will have fin tubes. The unit itself will occupy a space of 49 × 53 ft between building columns and will extend approximately 90 ft to the top of the air heaters, the mills being located at the basement level.

It is anticipated that this high-pressure extension to the station will be ready for operation late in 1941.

Combustion of Waste-Wood Products*

By H. W. BEECHER[1] and R. D. WATT[2]

A review of the economics and availability of hog fuel, with particular reference to the Northwest, its thermal value and combustion characteristics and the types of furnaces generally employed for burning it. The observation is made that the relatively low cost of this fuel has not encouraged engineering research that might involve greater capital investment leading to its more efficient use.

THE waste products resulting from the production of lumber, plywood or cellulose for conversion into pulp are available as fuel. Manifestly, any portion of the log which can be economically converted into more valuable material should neither be classified as waste wood nor used for fuel. Sawdust and shavings can be handled for burning without further processing. Slabs, edgings, trimmings and other waste products require further size reduction to prepare them for rapid combustion, easy transportation and convenient handling. Such material is usually processed by a mechanical masticator, commonly known as a "hog." The product so obtained, together with sawdust and shavings, forms a mixed fuel called "hog fuel."

Economic Aspects

Mill production cost for the hogged portion of fuel is from 10 cents to 15 cents per unit, largely made up of power, operation and maintenance expenses of the hogging equipment. Prices charged for hog fuel usually vary from 50 cents to $1.50 per unit at the producing mill, depending upon the supply and demand and not upon the cost of production.

The large volume and weight of hog fuel per available Btu makes the transportation cost loom large in the total cost to the consumer. The economical marketing zone is limited by transportation costs. Frequently, greater cost is involved in transportation and handling than the actual price paid by the consumer to the producer at the point of manufacture.

The high moisture content of hog fuel materially reduces the obtainable thermal efficiencies of boiler plants, as compared with the efficiencies secured with other fuels. This necessitates comparison of hog fuel with other fuels on the basis of their relative costs per available Btu. Many consumers of hog fuel pay as little as 50 cents a unit, delivered. In other Northwest plants the hog-fuel cost reaches $3.50 per unit. A unit of hog-fuel measurement occupies 200 cu ft. An average unit of hog fuel will contain approximately 20,000,000 Btu. Boiler-plant efficiencies with hog fuel vary, depending upon the type of installation, the percentage of rating at which the boiler plant operates and whether air heaters are installed for recovery of additional heat from the boiler gases. These efficiencies vary from 45 per cent on the poorer installations to 65 per cent on the more modern and better equipped boiler plants.

To indicate the general low cost of hog fuel for the production of steam, it may be noted that with 60 per cent efficiency the available heat per average unit would be 12,000,000 Btu. At a cost of $1 per unit, the corresponding cost of steam production would be $8^1/_3$ cents per million Btu input. With fuel oil costing $1 per bbl and with 83 per cent boiler efficiency, the heat available in steam would be 5,200,000 Btu per bbl and the corresponding cost per million Btu input would be 19 cents. With coal having a heat value of 12,500 Btu per lb and costing $4 per ton and with an average boiler efficiency of 80 per cent, the corresponding cost of steam per million Btu input would be 20 cents. This comparison indicates that, in so far as a consideration of the combustion of hog fuel is concerned, a low-cost fuel is involved which has not yet encouraged the engineering research or the capital investment which would be warranted were it a higher-priced commodity.

Availability of Waste-Wood Fuel

An analysis of the total volume of 1,354,000,000 cu ft of sound wood in the logs showed that approximately 67 per cent (911,000,000 cu ft) was converted into green, rough-sawed lumber; nearly 19 per cent (261,000,000 cu ft) became slabs, edgings and trimmings; the balance, 14 per cent (191,000,000 cu ft), was sawdust. In addition to the sound wood there was approximately 167,000,000-cu ft solid measure of bark. This indicates that of the solid-wood material, inclusive of bark represented by the logs as delivered to the sawmill, 41 per cent is so-called "waste" and available for fuel. It is reasonable to assume that the material left in the forest, such as tops and branches that cannot be economically handled, is sufficient to make up the difference between 41 and 50 per cent, and to state broadly that, of the wood content of the average tree as logged and utilized in the Northwest, less than 50 per cent is converted into lumber and its allied products. The

* Excerpts from a paper presented at the Fall Meeting of the A.S.M.E. at Spokane, Wash., September 3–6, 1940.
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balance is an economic waste except for the value recovered as fuel.

While there may be some slight improvement in utilization, the greatest encouragement for the conservationist comes from the possibility of chemically treating selected portions of existing waste for production of cellulose products. Another field is fermentation and production of alcohol. A third utilization process is destructive distillation which would give charcoal, numerous by-products and some steam for power production by burning the gas after cleansing of by-products. All these possibilities justify the statement that, as long as sawmills in the Northwest are operating at reasonable capacities, hog fuel in abundance will be available for industrial use.

There will remain a wide price range to the consumer at his place of use. As a consumer goes farther afield to secure his fuel in competition with other consumers, the more fortunately located sawmills with lower transportation costs will raise their prices and derive greater profits. Greater consuming markets will advance the prices so that small mills, beyond the zone of economical transportation at present fuel prices, will be provided with a market for materials now destroyed in refuse burners. Such destruction, while wasteful, is necessary without a market for this material.

Where water transportation is available without rehandling, it is by far the cheapest method of moving hog fuel. Hog fuel is towed on barges as great a distance as 350 miles.

In certain localities, transportation must be by rail in specially fitted cars. Tariffs are either per car or per 100 lb weight. In either case it is desirable to have car contents a maximum. Cars are constructed with sides extending to maximum clearance height for the division in which they are to operate. A 50-ft car may be constructed to handle 35 units. However, most cars carry from 20 to 30 units.

The use of trucks for hog-fuel transportation is increasing. Special bodies carry from 4 to 6 units where regulations will permit. They load by gravity from bins, or conveyers, and are fitted with power dumps by which they are discharged into hoppers.

In rare cases, the hog-fuel-producing plant is near the consuming market and belt conveyors can be used.

Heating Value

The high percentage of oxygen in wood reduces the heat content per pound as it is combined with carbon and hydrogen to form carbohydrates and, therefore, the total heat of combination of the combustibles is not all available. The manner in which these three elements are combined is not definitely known and the use of Dulong's formula, as applied to the ultimate analysis of wood, will not result in a Btu value corresponding to that obtained from calorimetric determinations. Hog fuel as normally delivered to the furnaces contains a high percentage of moisture. A portion of this is extraneous moisture, either resulting from wet logs, water lubrication of saws or rain when fuel has been exposed. Most of the moisture, however, is in the cellular structure of the wood. The fuel, as received from the average sawmill, contains material with high moisture content from the sap wood of the log, material with a medium moisture content from the heartwood and material with comparatively low moisture content from wood dried in commercial dry kilns.

In computing combustion results, moisture determinations are reported as the percentage of the total weight of wood and moisture that is represented by moisture. This means that fuel containing 50 per cent moisture contains 1 lb of water per lb of dry fuel. With the high oxygen content of wood there would be $1^3/_4$ lb of water per pound of combustible. If the oxygen were combined with the hydrogen, as assumed in Dulong's formula, 50 per cent moisture in the fuel would correspond to approximately 2 lb of water per pound of ununited or available combustible. It is interesting to note, when the moisture content is increased from 50 to 60 per cent, the weight of moisture in the fuel is increased from 1 to $1^1/_2$ lb per pound of dry fuel. The hog fuel used in industrial plants of the Northwest will average from 25 per cent moisture, when principally kiln-dried material, to from 57 to 60 per cent moisture when largely green hemlock.

All species of wood considered herein have approximately the same heating value on a bone-dry basis which will average 8900 Btu per lb of dry wood. However, some species are better fuels than others. Hemlock is not as good as fir. Spruce is better than hemlock but poorer than fir. Cedar is a light fuel and requires a specially designed furnace for good results. Hemlock, as ordinarily available as fuel, has a high moisture content and does not readily part with its moisture. At least 20 to 25 per cent more capacity can be obtained from given furnaces and combustion chambers with fir fuel having about 45 per cent moisture content than with hemlock having 57 per cent moisture.

The heating value of stored hog fuel varies with the time in storage. Storage of hog fuel in the open decreases the available Btu faster than storage under cover. This loss of heating value is attributed to slow oxidation which takes place at low temperatures. Cultures have been made from samples of hog fuel after storage over a considerable period which show an indication of molds and other wood-destroying fungi. These reactions are exothermic and the heat is lost. A typical ultimate analysis of wood is as follows:

	Per cent
Carbon	50.31
Hydrogen	6.20
Oxygen	43.08
Nitrogen	0.04
Ash	0.37

Combustion of Hog Fuel

Nearly 45 per cent of the dry weight of wood, independent of the species, is oxygen. The hydrogen-to-carbon ratio in wood is of the same order as in oil and, therefore, for the same excess air, the percentage of water vapor as compared to dry gases will be approximately the same for these two fuels. Coals, as a rule, have much lower hydrogen-to-carbon ratios and, therefore, give combustion gases containing lower percentages of moisture. The heating value of the fixed carbon in wood fuel amounts to from 15 to 20 per cent of the total heat in the fuel. The high moisture and volatile contents of hog fuel delay combustion which proceeds as follows:

1. The driving off of the moisture content and raising the wood to a temperature at which volatiles will be driven off;

2. The actual distillation of volatiles;

3. The combustion of the fixed carbon.

The high oxygen content of wood with its low nitrogen content reduces the percentage of nitrogen in hog-fuel flue gas. Coal of typical analysis, if completely burned without excess air, would produce $18^1/_2$ per cent CO_2 in the combustion gases. Similarly, oil of typical analysis, if burned without excess air, would produce $15^1/_2$ per cent CO_2. Wood of the typical analysis quoted will give, if completely burned without excess air, approximately 20 per cent CO_2.

The wood itself contains but little noncombustible in the form of ash. However, hog fuel as normally fired may carry with it appreciable quantities of ash-forming material in the nature of extraneous matter embedded in the bark or wood fibers and not removed in preparation, transportation and handling. This may consist of small pebbles, and sand shells. Logs which have been transported in salt water give off gases containing salt fume which assists in lowering the fusion temperature of the noncombustible and accelerates the deposit of slag on boiler tubes.

With deep fuel beds, most of the fixed carbon, undoubtedly, leaves the fuel bed as carbon monoxide where it unites with additional oxygen to burn to the dioxide. The incandescent carbon adjacent to the grates burns to the dioxide and then, in passing further through the incandescent carbon, is reduced to the monoxide.

In the cellular type of furnace, it is important to provide secondary or overdraft air. The conical pile of fuel is too thick, except around its edges, to pass the necessary air for rapid combustion. The closing of secondary-air admission ports, resulting from too thick a fuel bed, is quickly evident in the smoking of furnaces. The admission of overdraft air through the grates in the front of the furnace with little resistance to the passage of such air decreases the negative pressure in the furnace and lowers the required average draft throughout the setting. Any decrease in required draft is desirable to avoid infiltration in the convection sections where excess air decreases the efficiency of the boiler.

The standard method of feeding fuel to flat-grate cellular-type furnaces is through feed-hole openings located in the furnace roof, the fuel being transported to the furnace through chutes. Reasonable precautions are necessary to limit the amount of air entering the furnace through these chutes. Any air so admitted decreases the air to the preheater and also results in furnace stratification. In spite of such precautions, the falling fuel produces an injector action and entrains considerable quantities of air.

Design of Furnaces

The problems of proper combustion of hog fuel are greatly increased by the necessity of providing furnaces suitably designed for fuels varying in size from dust to pieces having 3 to 5 cu in. of content and for fuels of variable moisture content. Frequently, slugs of dry and highly combustible fuel are followed by slugs of wet fuel which form a damp blanket on the fuel pile. In the case of hopper-fired sloping-grate furnaces, one side of a hopper may contain dry fuel and the other side, wet fuel.

The designer must provide furnaces to handle properly the wet fuel and, at the same time, not to punish unduly refractories during the periods in which only dry fuel is fed. To produce the best average combustion conditions, much study has been given to the use of sloping-grate furnaces where the fuel is admitted in a comparatively thin and uniform layer over a drying hearth, in which portion of the furnace reflected heat is utilized to drive off the moisture and start the distillation process necessary before the fixed-carbon content of the fuel can be ignited. Following this section of the furnace, the fuel flows over grates and, as the volatile content is driven off, combustion of the fixed carbon is maintained by the air passing through the grates and fuel bed.

Sloping versus Flat Grates

Theoretically, such furnaces would be preferred to flat-grate, conical-pile furnaces with which by far the greatest number of hog-fuel-fired boilers are equipped. Practically, difficulties are encountered with sloping-grate furnaces caused by:

1. The fuel not being uniform in size and, therefore, containing streaks, or pockets, of greater density than adjacent areas, leads to the formation of blowholes through the fuel.

2. The fuel, not being of uniform moisture content, leads to the formation of areas in which distillation and ignition proceed more rapidly than in adjacent areas, thus resulting in the same formation of blowholes.

With a fuel as light as wood, particularly after the moisture and volatiles have been driven off, leaving charcoal cinders, these blowholes lift the cinders from the grates, depositing them at the foot of the sloping section and, in their formation, prevent the fuel from above the blowhole cascading to cover the hole. If too great ashpit pressures are used, this formation of blowholes is accentuated.

The accumulated charcoal cinders at the toe of the sloping grates afford such high resistance to the passage of air that insufficient air passes through this material. This prevents the combustion at the toe of the grate proceeding with sufficient rapidity to obtain high ratings per square foot of grate area, as the limiting rate for inflow of fuel over the drying-hearth section is the rate at which the fixed carbon can be consumed at the lower end of the grate. Even though sloping-grate furnaces have been tried in the Northwest with 15 and 16 ft of total length, obtainable capacities per foot of width of furnace have been less than those possible with well-designed furnaces of the so-called cellular type. As a result of the greater capacity obtainable in the latter furnace, most of the installations made in recent years have been of this design.

It is possible that extremely long sloping furnaces with special means for controlling the rate of feed and for cleaning the accumulated slag at the toe of the grate, with controlled and zoned air supply, could be developed to give results comparable with those obtained with a flat furnace. Such an installation would involve capital expenditures which do not appear to be commercially justified, as they could not improve materially upon the efficiencies obtained with the present flat cellular-

type furnace. An advantage of the cellular type of furnace is the ability to operate a boiler at reduced rating while burning down and cleaning the slag from the grates in one of the multiple cells. Cell-type furnaces are constructed with widths for individual cells ranging from $6^1/_2$ to $8^1/_2$ ft, which appear to be the economical limits of conical piles to be covered by single feed holes.

The combustion-chamber volume, gas-travel length before convection surfaces and the cross-sectional area of combustion space are related and important in hog-fuel combustion. In comparable installations, the gas weights with hog fuel are approximately 1.7 times the gas weights with oil, and approximately 1.25 times the gas weights with coal. This increased gas weight results in lower combustion-chamber temperatures which are further reduced by the high moisture content of the hog-fuel gases. The decreased temperature does not entirely offset the increased gas weights and larger cross-sectional areas are required when burning hog fuel to give comparable velocities in the combustion space. These factors make it essential to provide larger combustion spaces with hog fuel than with other fuels.

Use of Preheated Air

With the modern boiler installation, the increased capacity obtainable with preheated air has been largely responsible for the installation of preheaters rather than any gain in efficiency resulting from their use. When a preheater installation is charged with the extra capital, operating and power costs, made necessary by the installation of forced- and induced-draft fans and the necessary gas and air ducts, the low cost per Btu of the fuel precludes the justification of air preheaters on a strictly fuel-saving basis. With hog fuel it is impossible to obtain as low exit-gas temperatures from air preheaters as with other fuels. The high exit-gas temperatures, in part, result from the fact that only 75 to 80 per cent of the air required for combustion can be passed through the air preheater.

With the general introduction of water-cooled combustion chambers in an endeavor to reduce brickwork maintenance, the addition of the preheater has been found desirable in order to decrease the size of the combustion chamber and the length of gas travel necessary from the furnaces to the convection surfaces.

The use of preheaters has made it necessary to use water-cooled grates to avoid excessive grate maintenance. Water-cooled grates have also proved desirable to facilitate grate cleaning. The slag formed from the foreign matter brought in with the fuel does not adhere tenaciously to the water-cooled grates; whereas, with uncooled grates, it is removed with difficulty. Several designs of water-cooled grates have been developed for this service. The heat absorbed in the cooling water is low-potential heat and must be subtracted from the heat available for the production of steam. In many installations the heat obtained in grate-cooling water is used to heat condensate, or makeup water, in this manner supplanting heat which would otherwise be supplied by bled steam which had produced power or by the exhaust from noncondensing auxiliaries. It is, therefore, important in the design and construction of water-cooled grates to provide an arrangement for cooling which will extend the life of the grate, provide for easy cleaning and, at the same time, extract from the grates and from the preheated air passing through them a minimum amount of this low-potential heat.

Driers

Hog-fuel driers offer attractive potential savings to the power-plant operator. The flue gas leaving an air heater at approximately 500 F contains sufficient heat to remove about $^1/_2$ lb of water per lb of dry wood without dropping the temperature of the gas so low that condensation difficulties will arise. In addition to the savings, the drying of hog fuel gives considerably increased capacity per square foot of grate.

Although hog-fuel driers seemingly have a broad field, the volume of fuel to be dried per available Btu makes necessary a drier of such large physical dimensions that the fixed charges and the operating and maintenance expenses make it difficult to justify the investment.

Several different types of hog-fuel driers have been proposed and tried in this country and abroad, but the authors do not know of any design which has proved completely satisfactory. There is a definite field for a satisfactory hog-fuel drier, but until all of the mechanical difficulties with the prevailing designs can be successfully solved their use will not become extensive.

Cinder Nuisance

The cinder nuisance from hog-fuel-burning plants has increased with higher firing rates required in the modern high-duty boilers equipped with forced- and induced-draft fans and air preheaters. Modern hog-fuel-burning plants are providing either mechanical separators or flue-gas washers for cinder removal. Starting with the use of large single-cyclone dust separators, with relatively poor efficiencies on the fine light cinder particles, the necessity of securing better cinder elimination has led to a trial of various designs of mechanical devices and to the development of special wet gas washers.

Boiler Capacities

Tandem furnaces do not operate as well as those with a single feed hole per cell. Any boiler should have a minimum of two cells to permit carrying partial load during grate-cleaning periods, while three cells permit greater loads during such periods.

Numerous factors affect the capacity obtained from hog-fuel-fired boilers. The following table is intended to indicate in a general way what capacity should be expected from a well-designed furnace cell of the general dimensions used in modern installations in the Northwest. Values are given for cells with and without preheat and with good fuels of different moisture content:

Moisture in Fuel, Per cent	Btu Input per Sq Ft of Grate Area — Without Preheat	With Preheat
40	680,000	850,000
48	550,000	690,000
56	400,000	500,000

Caution should be exercised in the use of the capacities tabulated as they reflect what can be accomplished under good conditions and in properly designed furnaces.

Sizing of Coal for Chain- or Traveling-Grate Stokers

By WALTER WOOD

Combustion Engineering Company, Inc.

The importance of selecting coal with respect to its sizing when it is to be burned on forced-draft chain- and traveling-grate stokers is a matter that is often given too little consideration. This seems especially true with reference to the free-burning bituminous coals, but sizing is also an important factor in the selection of anthracite for use on a traveling-grate stoker. The effect which sizing has in burning both these types of fuels is discussed here. Reference is also made to the tempering of the midwestern free-burning coals.

DURING the last ten to fifteen years the majority of furnaces that have been built for forced-draft chain- and traveling-grate stokers have been of the rear-arch type, whereas earlier furnaces were of the front-arch type. The improvement in furnace arrangement has done a great deal to offset the troubles that were formerly experienced with poorly sized anthracite. Decided improvement has also been made in the mine preparation of the steam sizes of anthracite which has done much to increase capacity and efficiency of stoker operation.

The sizes of anthracite commonly used on forced-draft traveling-grate stokers are:

- No. 2 buckwheat, sometimes referred to as rice, passing a $^3/_8$-in. and lying on a $^3/_{16}$-in. round screen.
- No. 3 buckwheat, sometimes referred to as barley, passing a $^3/_{16}$-in. and lying on a $^3/_{32}$-in. round screen.
- No. 4 buckwheat which passes through a $^3/_{32}$-in. and lies on a $^3/_{64}$-in. screen.

Stokers installed in front-arch furnaces burn No. 2 and No. 3 buckwheat almost exclusively. The relatively small amount of No. 4 buckwheat that is burned in front-arch furnaces gives poor results both as to capacity and efficiency. In this type of furnace, ignition of the fuel is often slow and difficult to maintain; yet the fuel must be burned rapidly for best results, once it is well ignited. The air pressures under the fuel bed necessary to secure rapid and complete burning are high enough to lift much of this very fine coal off the grate. Some of the fines in suspension drop back on the fuel bed and can be burned, but much is deposited on the rear of the grate beyond the last air zone and is lost. Other fines are carried directly into the ashpit, and a large amount goes out of the furnace in the gas stream.

The carbon lost in this way depends principally on the rate of burning of the coal. It is not possible to give exact values as to this loss, because of differences in furnace design and in the amount of undersize in the coal, but it is safe to say that the loss may be as high as 15 to 18 per cent of the total coal fired at the maximum rates of burning. At normal combustion rates the carbon lost in per cent of fuel burned in front-arch furnaces using No. 4 buckwheat is from $2^1/_2$ to 3 points higher when the undersize through a $^3/_{64}$-in. screen is 25 per cent than it is if the undersize is only 15 per cent. It is evident, therefore, that even in the relatively few cases where No. 4 buckwheat is burned in front-arch furnaces, it is important to specify the percentage of undersize when purchasing this fuel.

No. 3 buckwheat is the size of anthracite most commonly used on forced-draft traveling-grate stokers in front-arch furnaces. It is more easily ignited than either No. 2 or No. 4 buckwheat, and if properly sized

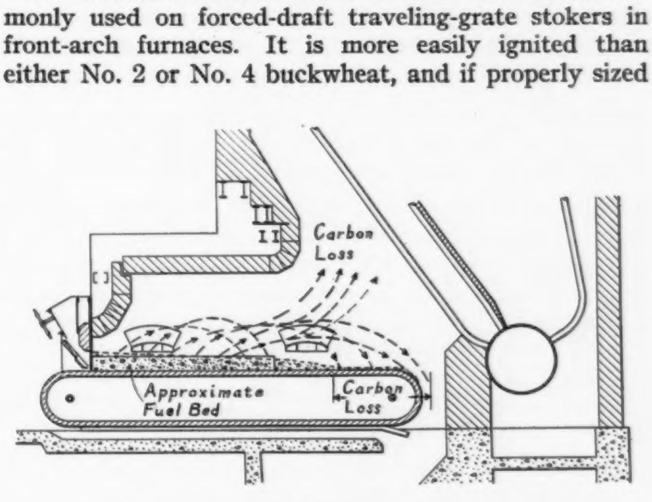


Fig. 1—Path of fines with front-arch stoker

the losses incurred in burning it can be kept fairly low. Sufficient air can be used as soon as the fuel is well ignited to burn the coal without likelihood of carbon monoxide loss; and, if the amount of undersize in the coal is not in excess of about 20 per cent, the carbon loss may not be unreasonably high. At what might be regarded as maximum combustion rates with No. 3 buckwheat burned in front-arch furnaces, depending on stoker length and furnace design, the carbon loss will lie between about 16 per cent for coal with 20 per cent undersize and about 20 per cent when the undersize is 30 per cent.

All of the steam sizes of anthracite can be burned at higher rates of combustion with much lower carbon

losses if the stoker is set in a well-designed rear-arch furnace. The rate of increase in carbon loss in rear-arch furnaces, with increase in combustion rate, is much lower than in the case of front-arch furnaces. The advantage which the rear-arch furnace has over the front-arch type with regard to carbon loss is that the fines lifted by the air from the fuel bed are carried away

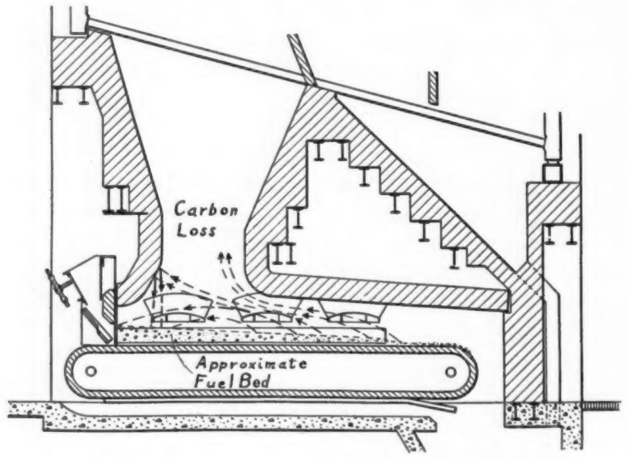


Fig. 2—Path of fines with rear-arch stoker

from the ashpit, toward the front of the furnace. Most of these fines drop onto the fuel bed where they are burned. Some fines are lost by being carried out by the gases. That loss is proportionate to the undersize.

When operating a rear-arch furnace at what might be regarded as a high combustion rate with No. 3 buckwheat having 20 per cent undersize, the carbon loss is on the order of 10 per cent of the fuel fired, whereas a loss at the same combustion rate will likely be 15 per cent if there is 30 per cent undersize in the fuel. In general, the same relation holds true for No. 2 and No. 4 but the carbon losses are smaller with No. 2 and greater with No. 4. It is important, therefore, that the amount of undersize in any of the steam sizes of anthracite should be known when buying any of these coals for use on stokers of the forced-draft chain- and traveling-grate types, whether for front-arch or rear-arch furnaces.

The selection of free-burning midwest bituminous coals as to sizing for use on stokers of these types has probably been influenced more by custom or habit than by the results of careful study of operation with these coals of different sizing. For years, about the only specification as to size of free-burning coal for forced-draft chain-grate stokers has been that the coal shall pass through a $1^1/_4$-in. round-mesh screen. Sometimes the amount of fines or undersize has been mentioned, but only in rather exceptional cases. There seems to be no generally accepted idea as to the definition of the term undersize as applied to free-burning coals for these stokers. In the absence of a standard screen for determining undersize or fines in these coals, it is suggested that a $^3/_{32}$-in. round-mesh screen be employed.

The amount of fines in free-burning bituminous coals burned on chain-grate stokers is not as important a factor as the maximum sizing, but it does govern to a considerable extent both the capacity and efficiency that can be obtained. The percentage of intermediate sizes also affects the manner in which the fuel will burn.

A hard run-of-mine bituminous coal crushed to pass a $1^1/_2$-in. screen will contain a rather small amount of fines and the percentage of intermediate sizes will be correspondingly high. On the other hand, when a very friable coal is crushed and screened to the same maximum sizing the percentage of fines will be relatively high.

When the maximum sizing of free-burning midwest coals burned on chain- and traveling-grate stokers is as much as $1^1/_4$ in., ignition is very likely to take place slowly. The delayed ignition makes it impossible to reach a maximum combustion rate until the fuel has been carried well into the furnace; and the higher the percentage of intermediate sizings, the more this difficulty is aggravated. In a fuel bed composed principally of large and medium sized lumps there is not sufficient contact between ignited surfaces to insure combustion except at a very low rate. There must be enough fines uniformally mixed with the larger lumps to provide a fairly compact bed in order to get prompt ignition and rapid combustion. The amount of fines should not be too great, however, otherwise the air for combustion may produce holes in the fuel bed.

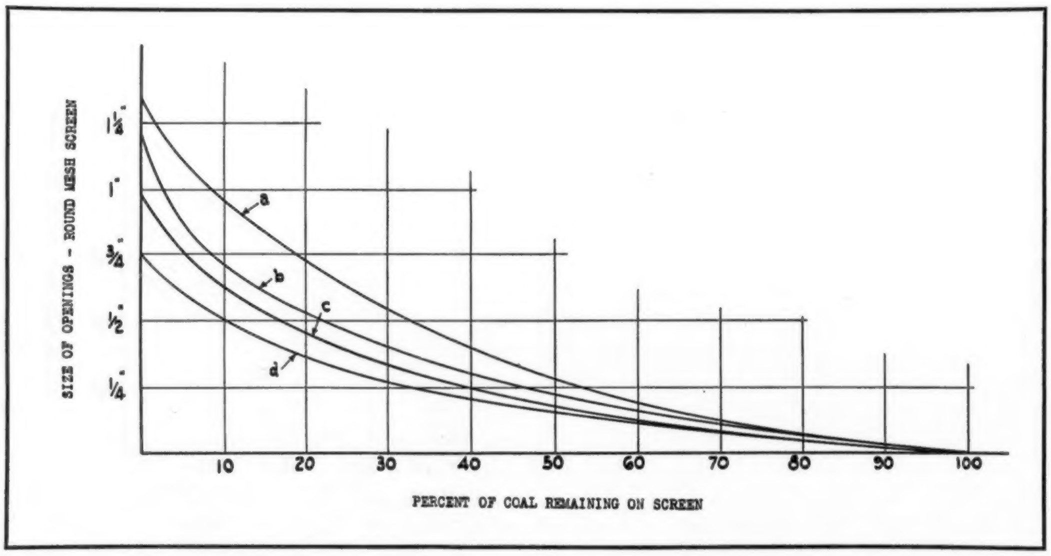


Fig. 3—Screen sizing for free-burning bituminous coals

Experience indicates that coal sized through $1^1/_4$-in. screen is too large for most satisfactory operation. The largest lumps frequently do not burn out completely before they reach the rear of the stoker, and a high carbon loss results. Also, the lumps, if not evenly distributed, will allow too much air to pass through the fuel bed, and the CO_2 will be low. Most of the free-burning coals give best results if the maximum sizing is about $^3/_4$ in. Coals of that size ignite quickly in most furnaces and ignition penetrates the fuel bed quite rapidly, so maximum rates of burning can be expected by the time the fuel has been carried only a short distance into the furnace.

The amount of fines in fuel prepared through a $^3/_4$-in. screen should be on the order of 20 or 25 per cent through a $^3/_{32}$-in. screen. The larger lumps in such coals burn out rapidly and carbon loss in the ashpit is low. At the same time the fuel bed is sufficiently compact to insure high CO_2. The curves shown in Fig. 3 represent actual screen sizings of free-burning bituminous coals burned on tests. The coal represented by curve *a* contained so much of the larger sizes that ignition and combustion were very slow. The lumps did not burn out well and the carbon loss in the ashpit was high. The CO_2 was low due to the free passage of air even through a 7-in. fuel bed. Coals represented by curves *b* and *c* gave better results. The lowest losses in carbon in the ashpit and in excess air were obtained with the coal represented by curve *d*. Ignition with that coal occurred at the regulating gate and penetration through the fuel bed was very rapid.

The uniformity of distribution of the lumps and fines in a fuel bed is very important. If the coal is discharged into the stoker hopper through an ordinary spout in a fixed position the lumps separate from the fines and move to the ends of the stoker hopper. The result is a fuel bed of lumps along the side walls and a more or less uniform mixture of the smaller sizes toward the middle of the stoker. The lumps will not ignite unless the air pressure is kept low, and therefore the middle portion of the fuel bed will not burn completely. Coal supplied to the stoker hopper from a traveling larry or through a spout which continually moves from one end of the stoker hopper to the other insures a uniform fuel bed.

The free burning coals of the Middle West represent by far the largest as well as the most satisfactory supply of bituminous coals for chain- and traveling-grate stokers. These coals will all coke to some extent. If they are burned with only their normal moisture content, there will be excessive loss of carbon in the ashpit refuse. Moisture must be added to insure minimum carbon loss. Most of these coals require a total moisture content of about 14 to 15 per cent, but the moisture should be even higher in many of the coals found west of the Mississippi. Iowa coals, for example, generally need a total moisture content of about 18 per cent for satisfactory burning. The adding of moisture, called "tempering," should be done long enough before the coal reaches the stoker to insure penetration of the lumps by water. A period of about twenty-four hours should be allowed for thorough tempering if possible. Water added to the coal by sprays in the stoker hopper does very little to improve the burning of these coals. Better results can be attained by the use of steam applied through a number of small pipes located in the stoker hopper, but tempering with water at least a day before the coal is to be burned is preferred.

Steam Generation Trends in 1940

A partial survey of units ordered during the current year, covering thirty-five for central-station service and eighty for industrial plant use. These are analyzed as to capacities, steam conditions and methods of firing.

IN VIEW of present activity in the power plant field the question is sometimes asked as to whether trends of the last few years are being followed or whether certain departures from these trends are being made to meet prevailing exigencies. Such a question can best be answered by scanning some of the orders placed for steam generating equipment during the current year. In the absence of a complete record of such units, a partial list has been compiled, comprising thirty-five central-station units, aggregating approximately fourteen million pounds per hour of steam generating capacity, and eighty units for industrial power plants, totaling nearly five million pounds per hour, which may serve as a useful indication.

Central Station Units

The utility units here considered range in output from 60,000 lb to a million pounds of steam per hour, fourteen being for 500,000 lb or over. Although one large unit will operate at only 300 lb pressure to conform to existing steam conditions in the station of which it is an extension, five others will employ pressures between 500 and 750 lb, sixteen between 750 and 1000 lb, eleven between 1000 and 1500 lb and two over 1500 lb, the highest of these being 2000 lb. About half of these units are for topping installations and the remainder will supply condensing units.

As concerns total steam temperatures, one will employ 690 F, nineteen will operate between 735 and 900 F and fifteen over 900 F, the maximum being 960 F. The reheat cycle will be used in two cases.

Pulverized coal predominates, with thirty of the units selected being so fired; whereas four will be stoker-fired and one will burn oil or gas. The largest of the stoker-fired units has a rated output of 185,000 lb of steam per hour. It should be mentioned, however, that many of those designed to burn pulverized coal will also be equipped to burn oil or gas, if desired. Twenty-two of the pulverized-coal-fired boilers will have dry-bottom furnaces and eight slagging bottoms. The latter are all high-capacity units. Their number, compared to that of the dry-bottom type, can hardly be taken as an indication of trend, as the selection of furnace type is largely governed by fuel, load characteristics and local conditions. All of the boilers considered in this survey are of the bent-tube type.

Therefore, as concerns utility units, there appears among recent orders, with one exception, to be little departure from trends of the past two or three years. Such departures as appear pertain mostly to certain details of design based on experience with earlier units of comparable size and type. There has been a slight increase in the total steam temperature over the prevailing high of a year ago, and the number of high capacity units appears to be increasing. While units for moderate steam pressures slightly outnumber those for very high pressures, there is no indication of a recession in pressures and in at least two cases they have been extended. Also, present oil prices and the uncertainty of supply, in view of possible future naval demands, seem to have been reflected in the greater number of coal-burning installations. The one outstanding example of a departure from previous years is the introduction of forced circulation into American practice.

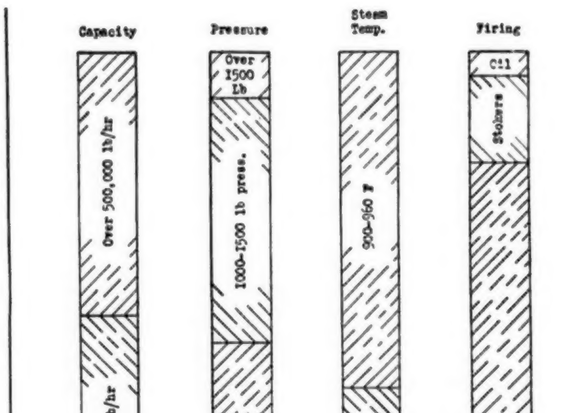


Fig. 1—Capacity range, pressures, total steam temperatures and methods of firing thirty-five central-station boilers ordered in 1940

NOTE: A number of these units, although designed to burn pulverized coal initially, will also be capable of burning either oil or gas if desired.

Industrial Plant Units

The industrial plant boilers cited cover numerous industries and range in capacity from 10,000 to 260,000 lb of steam per hour. Units larger than this are not included as they usually involve special conditions not representative of industrial requirements in general. Of the eighty boilers considered, twenty-nine are stoker-

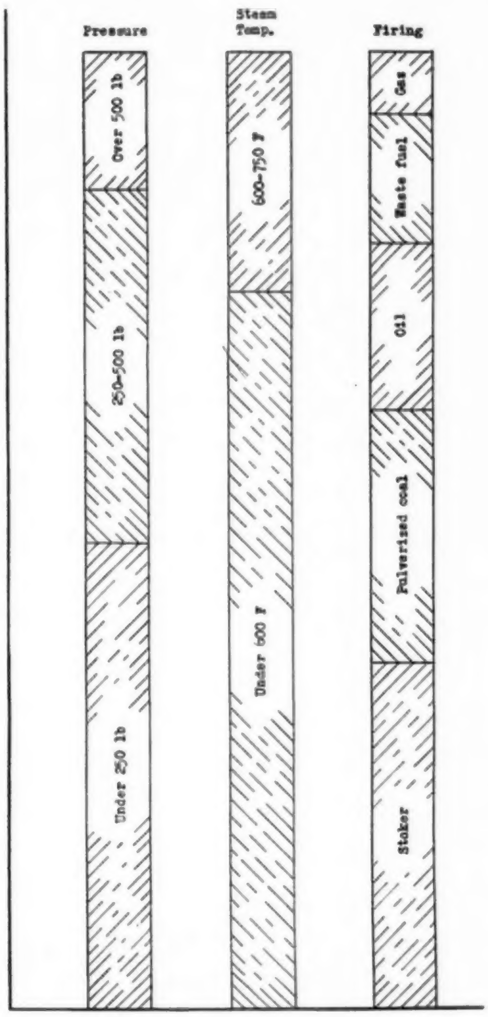


Fig. 2—Ranges in pressure and total steam temperature and methods of firing employed by eighty industrial plant boilers ordered in 1940

NOTE: This does not cover a very large number of smaller units for stoker firing.

fired, twenty-one pulverized-coal fired, fourteen oil-fired, five are fired by natural gas, and the remainder will burn waste fuels which form a by-product of the several industries served. Those employing pulverized coal, in general, range in output from around 30,000 to 200,000 lb of steam per hour. Stokers predominate below 30,000 lb per hr although in some instances units of 50,000 to 85,000 lb per hr capacity are so fired.

Only eleven of these units will employ steam pressures above 500 lb; thirty will use between 250 and 500 lb, and the remainder, which represents about half of those considered, will operate under 250 lb. Steam temperatures in excess of 600 F are found in twenty cases, the highest being 750 F. As to boiler types, the two-drum bent-tube design predominates, although there are thirty-one of the three- and four-drum type and fifteen of the straight-tube box-header and sectional-header designs, principally where moderate steam pressures are involved.

Without knowing what ratio these eighty industrial power boilers bears to the total number purchased thus far during the present year, one is justified only in considering them as indicative of current practice in that field. For instance, the eleven units for pressures in excess of 500 lb, compared with the large number for pressures of 250 lb and under, does not imply a recession from higher pressures in the industrial power field. There always have been and will be many more plants whose size, load conditions and steam use properly fall within the low-pressure category than there are those which can economically use high pressures. In his paper on "Progress of Higher Pressures in the Industrial Plant," before the A.S.M.E. Spring Meeting at Worcester last May,[1] W. F. Ryan was able to account for eighty-five industrial plants employing pressures of 500 lb and over, but this survey covered installations that had been made during the last decade, whereas the present figures (which are not complete) apply only to the first half of the present year.

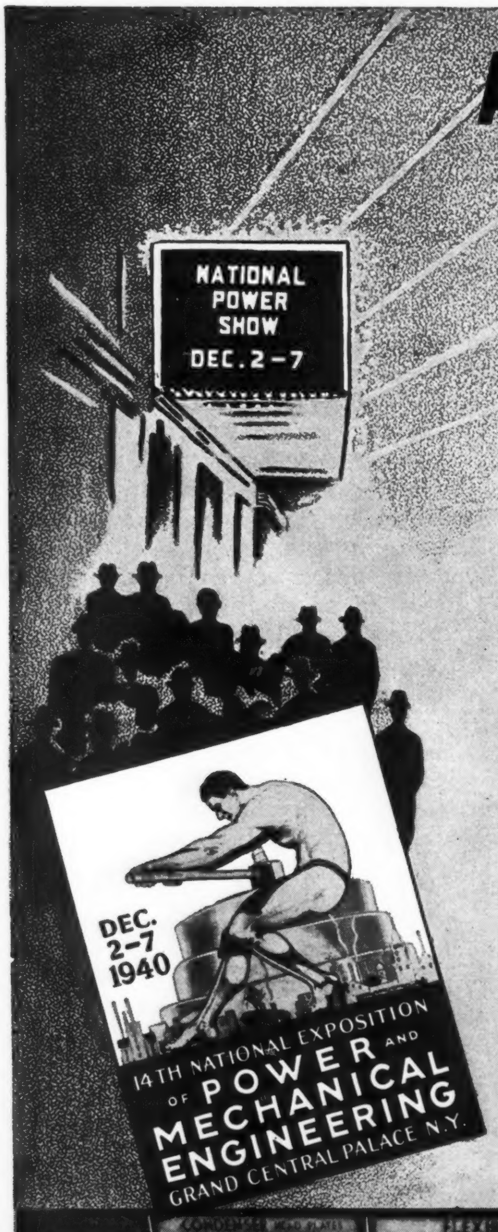
Experience of the numerous larger plants that already have adopted higher steam pressures, appears to have fully justified the selection, as several of these have been extended for the same steam conditions. On the other hand, except in the case of straight condensing installations which represent a minority in the industrial plant field, there is little reason to employ high steam temperatures.

Finally, it warrants reiterating that the foregoing presents only a partial picture of activity in the field during the present year, based on information readily available. Furthermore, no account is taken of the very large number of stokers being sold for existing boilers and new boilers of smaller capacity. It is believed, however, that this survey is sufficient to provide a cross-section of current trends, and will serve to indicate 1941 operating practice, inasmuch as most of these units will not go into service until next year, whereas those now being placed in service were ordered in 1939.

[1] See COMBUSTION, May 1940.

Date for Bituminous Coal Prices Extended

In the August issue of COMBUSTION were printed representative minimum bituminous coal prices as established by the Bituminous Coal Division of the Department of the Interior. At the time of going to press it was announced officially that these prices would become effective on September 3. Subsequently, however, on August 16 the Director of the Bituminous Coal Division issued an order postponing the effective date of these prices to October 1. This action resulted from an order issued by Secretary of the Interior, Ickes, extending to August 30 the time within which requests for review of the price schedules might be filed. These requests are now reported to be receiving consideration.



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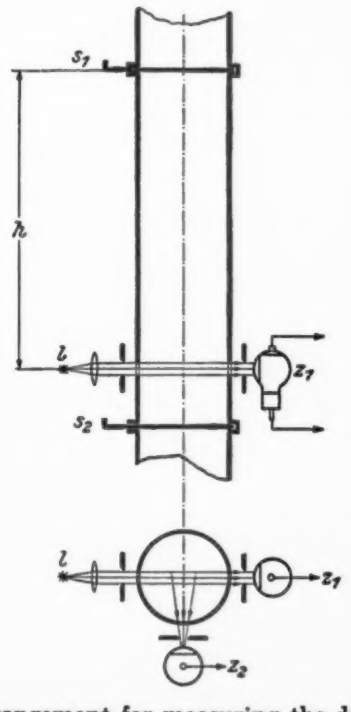
Determining Grain Distribution in Dust-Laden Air

VDI Zeitschrift, Vol. 84, No. 8, contains an article by K. Gösele which presents a new method for determining the grain distribution of a lean dust mixture in air or gas. Ordinarily, it is impossible to obtain a sufficient quantity of the dust for analysis by drawing off a given portion of such a mixture.

The principle of this method involves measuring the settling of the dust particles in a sealed space by light-electric means. Referring to the figure, the dust-laden air to be examined is passed through a vertical tube and



Schematic arrangement for measuring the dust distribution in air by light-electric means

h = height of fall
s_1 and s_2 = dampers
z_1 and z_2 = photoelectric cells
l = source of light

is segregated in a portion of tube by dampers s_1 and s_2. A source of illumination l sends a bundle of substantially parallel light rays across the tube through the dust-laden air.

A photoelectric cell z_1 measures the light strength J after having passed across the tube and thereby measures the weakening of the light by the dust-laden air. Since the smaller dust particles only slightly weaken the light and are therefore inaccurately measurable in this way, there is provided a second photoelectric cell z_2 for measuring the light which is deflected sideways. The change J in the strength of the light within time t is dependent upon the separation of the particles which fall from height h during this time. The grain size (a) may then be determined from the falling time by Stokes' Law.

Plotting the values of J in a diagram having grain size (a) as abscissa, one obtains the settling curve. The deflected light strength and the logarithm of the light weakening, for particles above 2 μ diameter, are more nearly proportional to the dust particle surface (in cm^2 of grain surface per m^3 of air) and not directly to the dust content (in mg/m^3). By differentiating the settling curve and multiplying this value by the value of the particle size, one can obtain the ordinates for the grain distribution line. By piecemeal planimeter measurements the line of material retained can be determined.

The light-electric method is applicable to particles having falling velocities of from 0.01 to 5 cm/sec, equivalent to particle sizes of from about 1 to 30 μ. Dust densities as low as 10 to 20 mg/m^3 may be measured. The shortcoming of the method lies in the fact that it provides accurate values only when the dust is uniformly distributed with regard to density and reflection of light.

When measuring large grain size zones it is disturbing that the sensitivity of the method is inversely dependent upon the grain size, i.e., fine particles are measured with greater accuracy and coarse particles with materially less accuracy.

Minimizing Standby Losses

F. J. Matthews in *Electrical Times* for July 25 writes that experience at a number of power stations in the past few years indicates that substantial economies can be effected in the operation of standby boilers which are held in reserve to meet load fluctuations.

Investigations show that radiation from the outside surfaces of a boiler is comparatively small during banked periods, and that the greater part of the heat losses encountered during such periods is interior radiation from hot brickwork and heat-transmitting surfaces. The chimney draws a substantial quantity of air through the furnace; the hot air is replaced by cold air; and the heat passes out to the atmosphere.

Heat losses of this character can be prevented by the provision of tight dampers which will effectively isolate the chimney from the furnace and boiler. Air flow through the unit during the standby period is then prevented and the furnace temperature can be maintained without banking the fires,[1] so that the boiler can be put on the line rapidly when required. In many instances, where such tight dampers have not been provided, repeated firing is necessary during the standby period to maintain boiler pressure to meet peak loads. The amount of fuel lost in banking is much more substantial

[1] NOTE: It should be borne in mind that a banked stoker-fired unit must maintain some draft through the furnace in order to allow inflammable distillates to escape, to prevent furnace gases from entering the boiler room and to sustain combustion. This is essential to prevent explosions, to prevent pollution of the boiler room and to minimize fuel losses.

than that required to maintain the furnace temperature and steam pressure. One design of tight damper, mentioned by the writer, consists of an asbestos curtain which is rolled into the flue ducts from a roller, tight sealing being insured by the flexible material and its pressure against the damper frame. Another tight damper consists of two sealing surfaces separated by an air space which is opened to the atmosphere when the damper is closed. In this position the boiler and furnace gases are held within the unit by the first sealing section while the natural draft caused by the chimney circulates outside air.

Plants which have installed these dampers report substantial improvement in efficiency over previous performance. One plant obtained an improvement of 3 per cent representing 300 tons annually. A smaller plant which was fired with chain-grate stokers bettered performance by 5 per cent or 200 tons per year, another plant saved 100 tons per year, a pulverized fuel fired plant increased its efficiency by $2^1/_2$ per cent, and another chain-grate stoker-fired plant reported 350 tons savings per year. In each instance these figures apply to boilers which are shut down for 12 to 14 hours daily; should they be shut down for longer periods it is apparent that even better efficiency improvement might be expected.

The application of tight dampers commonly reduces the time for starting up by 50 per cent or more. One factory reports reduced starting up time from $1^1/_4$ hr to $^1/_4$ hr; in another instance the time was reduced from 1 hr to $^1/_2$ hr. Two power stations attained 50 per cent reduction in starting up time, while a third station puts a standby boiler on the line in $^1/_4$ hr after $7^1/_2$ hr of idleness and also eliminates the use of $^1/_2$ ton of coal per day for banking.

It has been noticed, during the first hours of operation of units equipped with the improved dampers, that 15 to 18 deg higher superheat is available than was the case previously. This is attributed to the sustaining of uniform higher temperatures of brickwork around the superheater section. Maintenance charges have been reduced because of the smaller fluctuations in brickwork temperatures with correspondingly less wear and deterioration.

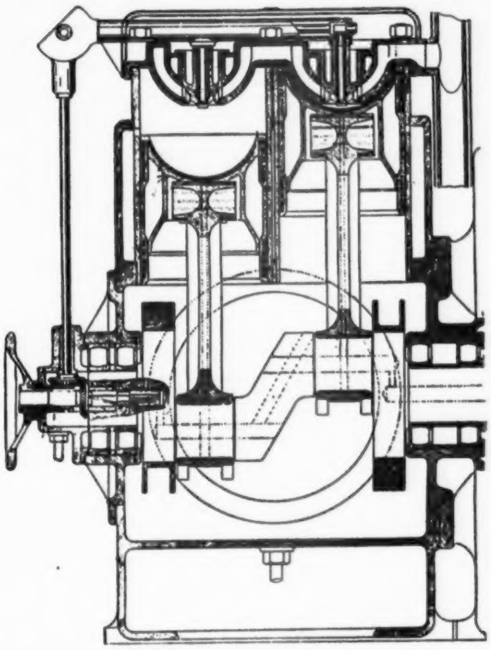
A Novel Steam Engine

In the August issue of *Engineering and Boiler House Review*, there is a description of a new steam engine employing a successful adaptation of the Lentz poppet-valve design and the uniflow principle. Sensational claims have been advanced by Dr. Lentz, the well-known Austrian engineer, regarding its steam economy and low unit weight.

The interesting features include camshaft-operated Lentz poppet valves, a cupped piston head which provides a minimum clearance and high compression ratio, together with the application of the uniflow principle for exhausting through ports in the cylinder wall. Without stating operating conditions, the designer claims to be able to guarantee a Rankine cycle efficiency ratio of 92 per cent, and a steam consumption of 6.6 lb of steam per ihp. With 842 F final steam temperature the steam consumption can be reduced to 5.5 lb per ihp.

A low unit weight of 6.6 lb per ihp is claimed as a result of the high speeds of 900 to 1200 rpm to be used in the

six- and the nine-cylinder engines which, it is intended, will be put into mass production. They will be designed for outputs ranging from 250 to 1250 ihp and from 375



Section through engine showing piston and valve chamber

to 1875 ihp, respectively. A special design to serve as a locomotive wheel drive is said to have achieved a unit weight of 2.2 lb per ihp.

The cylinder heads are steam jacketed, the hollow pistons are evacuated, and the steam supply pipe is shielded against radiation loss by placing it within a pipe of larger diameter and evacuating the annual space between the two. There is provision for an ample oil supply for lubricating and cooling purposes, and it is claimed that superheated steam up to 800 F can be used without incurring lubrication difficulties.

Velox Units in Holland

The Brown Boveri Review for April describes a recent installation of two 110,000 lb per hr Velox steam generators in the Velsen Power Station at Bloemendaal, Holland, which supplies power to the whole province of "Noordholland" with the exception of the cities of Amsterdam, Haarlem and Zaandam.

The new boilers, which are oil fired, supplement six 143,000-lb per hr boilers of standard design that have been in service for some time operating partly on coal and partly on blast-furnace gas from a neighboring steel plant. The deficiency of blast-furnace gas was responsible for the use of oil with the Velox units which handle only peakloads or furnish standby service. Normal designed steam conditions are 440 lb per sq in. and 800 F total steam temperature. The net weight of each unit with accessories is stated to be 72 tons or 1.44 lb per pound of steam.

Service test data on one of the units is given as follows:

Load, thousand lb per hr	35.2	57.2	82.5	110
Per cent of full load	32	52	75	100
Steam pressure, lb per sq in.	432	446	450	454
Steam temperature, F	703	718	750	784
Feedwater temperature, F	252	250	250	252
Efficiency of unit, per cent	87	90	91.5	92

Power Development in Turkey

In Combustion, November 1939, we reported in this department that plans were being put into effect for electrifying 35 more cities in Turkey. The *Electrical Times* (London), May 2, reports a contract, recently made public, between the Turkish Government institution known as Eti-Bank and the Metropolitan Vickers Engineering Co., Ltd., covering the electrification of a large and important area, including the construction of a power station at Catalgzi on the Anatolian coast of the Black Sea.

One of the main objects is to provide electric power for the modernization and expansion of the Turkish coal industry. The site of the power station was chosen for its proximity to the Zonguldah coal basin from which fuel for the station will be derived. The present contract includes the building and equipment of a 60,000-kw steam power station and the provision of switch gear, overhead lines and three substations for a transmission system. Provision is to be made for future extension of the program.

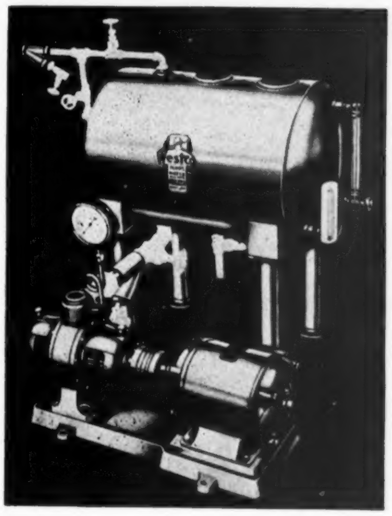
The initial generating plant will consist of three 20,000-kw turbine-generator sets to operate at 3000 rpm with Metrovick central-flow condensers and 4-stage feed-heating. The main units will generate at 11,000 volts and each set will include a 1500 kw house-service generator. A 375-kw diesel-engine set will be provided for standby service to the station auxiliaries.

This contract is one of the first results of an active trade policy designed to take a portion of the trade from Germany which has hitherto held a major portion of trade in the southeast corner of Europe.

NEW EQUIPMENT

Boiler Feed System

Micro-Westco, Inc. has recently brought out a new line of Westco automatic boiler return systems incorporating special features applicable to boilers up to 750 hp and 200 lb pressure. These systems are designed to return automatically all condensate to the boiler, to add makeup water, and to maintain a uniform water



level. The automatic control starts and stops the feed pump to maintain the water level in the boiler within a $^3/_4$-in. range, cuts off the fuel feed if the water should reach a dangerously low level, and can be wired to sound an alarm when the low cut-off point is reached. The control is fitted with a monel float, has a specially constructed bellows to eliminate cracking and is constructed with all operating parts isolated from the steam and hot water zone. Each combination is equipped with a Westco turbine-type pump having renewable liners and bearing adjustment for centering the impeller.

CO_2 Analyzer

For measuring the percentage of CO_2 in the flue gases of boiler furnaces, Bacharach Industrial Instrument Company offers a chemical type CO_2 analyzer, trade-named FYRITE. An important new feature is the gas sampling equipment which includes a primary flue filter with a replaceable filtering thimble.

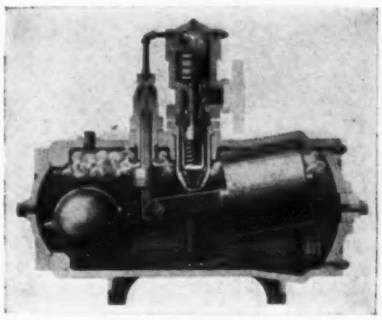
In operation, a flue-gas sample of known volume is pumped to the instrument by hand with a rubber bulb and trapped in the instrument automatically by detaching the sampling hose. The instrument is then turned upside down and back again to mix the gas sample with the absorbing reagent, which is said to be good for several hundred samplings before replacement is required. The suction created due to the complete absorption of the CO_2 pulls the absorbing fluid up an

amount equal to the CO_2 absorbed. Consequently, its level shows the percentage of CO_2 in the gas. The entire operation, including pumping of the gas, takes less than a minute.

High-Capacity, High-Pressure Drainage

The Cochrane Discharger here illustrated is essentially a positive-acting trap of specialized design to handle large quantities of condensate or carryover at relatively high pressures. At high differential pressures, the drainage of large quantities of condensate or boiler carryover presents a problem to usual types of drainage equipment. To insure tightness, an unbalanced value must be used. At the same time the discharge orifice must be large enought to drain slugs of water rapidly from steam purifiers or dry steam drums of priming boilers.

High operating power, unrestricted by pressure-capacity relationships is the ideal solution. The Cochrane Discharger, by means of a pilot valve, applies the steam



line pressure to open the discharge valve. Displacement weights of different densities move the pilot control valve. When the pilot valve has opened, steam is admitted to the top of the main valve piston, opening the discharge valve. Drains are discharged from the bottom of the body through the eduction pipe to the outlet. Pilot-control together with independence of priming are advantageous factors particularly in conjunction with pulsating pressures.

Typical applications are to boiler dry drums, steam purifiers and separators, and to steam mains.

Increasing Zeolite Softener Capacity

A new method of controlling the distribution and flow of water in a zeolite softener, known as the "Double Check Strainer System," has been announced by the Elgin Softener Corporation. The purpose of this system is primarily to prevent the loss of zeolite; to obtain better water distribution, thereby cleaning the bed more thoroughly; to increase the capacity of a water softener; and to permit the use of a larger amount of zeolite in a softener of given size.

The action of the double-check strainer used in the upper manifold is shown in the illustrations of the device. Illustration *A* shows the double-check strainer in the service position. The lower check valve *d* is closed by the flow of the water from the manifold, so that the water cannot pass through the strainer but must follow the course of the arrows, leaving through the open check valve *c* and spraying evenly over the zeolite bed. The slotted strainers in the lower manifold permit free flow of water to the service line, yet prevent the escape of zeolite mineral.

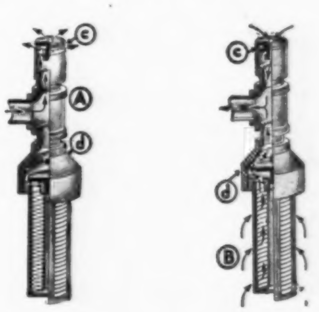


Illustration *B* shows the double-check strainer in the backwash position. Here the flow is reversed so that most of the backwash water passes through the strainers and around the now opened check valve, *d*, to the waste line. The openings in the upper check valve, *c*, carry a limited predetermined amount of the backwash water with the sediment to the drain, but the remaining backwash water flows through the strainers, gradually reducing the flow of water in that portion of the softener tank between the bottom of the strainers and the top of the strainers.

Silica Removal

The Permutit Company announces that it has perfected a commercial process for silica removal, thus preventing silica scale deposits in boilers, superheaters and turbines. The process makes use of the silica-absorbing property of metallic oxide

sludges. The water being treated is brought into the most intimate contact with high concentrations of sludge, thus assuring maximum silica removal. Operating costs are said to be low and soluble by-products are not formed. Therefore, the total dissolved solids in the feedwater is not increased but is usually reduced.

The process can be used in connection with both hot and cold lime-soda processes of water softening and is adaptable to either new or existing equipment.

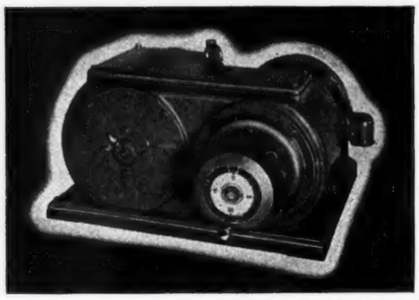
Smoke Indicator

Photoswitch Incorporated announces a new Photoswitch smoke alarm with densometer. It consists of a photoelectrically-controlled densometer and red signal light to indicate smoke densities above a predetermined value consistent with efficient combustion.

The photoelectric control and light source are mounted on opposite sides of the flue or breaching and are aligned through simple entrance tubes or windows so that the beam projected from the light source strikes the eye of the control. The densometer may be placed at any convenient location in the boiler room and is wired to the photoelectric control.

Speed Control and Indicator

Reeves Pulley Company announces a new, improved type of handwheel-speed indicator for use with the Reeves variable speed transmission, vari-speed motor pulley and motodrive. This equipment, known as the "Speedial," indicates speed



settings of the different units. The actual indication is a definite number of turns of the speed-shifting screw of the unit. For each full turn of the shifting screw, the Speedial, with indicator pointer attached, registers one point or degree on the circular scale.

It is offered as optional equipment, at extra cost, in place of the standard speed-control handwheel and indicator on the various Reeves drives, and is available for use on both new units and on units already in service.

While the standard Speedial is calibrated in turns of the shifting screw, space is available on the dial for the user to write, in pen or pencil, his own calibrations in whatever corresponding units he prefers. Blank dials can also be calibrated to the user's individual requirements.

Straight-Line Draft Gage

The Ellison Draft Gage Company announces a new Straight-Line Diafram Draft Gage which has a high power, free floating diaphragm, with a travel of from 0.25 to 0.6 in. and a displacement of at least 10 cu in. at 1 in. pressure. The location of the diaphragm on the bottom of

the gage casing, suspended through a ball joint from the knife edge eyelet bearing, is said to result in a perfectly mechanized diaphragm movement.

The diaphragm fabric consists of a fine mesh, pure silk cloth, coated with a pliable and airtight synthetic coating. The gages are made for plus, minus or differential pressures throughout a large range.